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SPACE POWERPLANT STARTUP AND TRANSIENT OPERATION SIMULATION WITH DIRECT-CONDENSING RADIATOR SYSTEM

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Technical Management
NASA Lewis Research Center
Space Power System Division
Nuclear Power Technology Branch
Cleveland, Ohio
Martin Gutstein

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Written by J. R. Hooper — J. R. Hooper, Sr. Anal. Engineer

Written and Approved by H. R. Kunz — H. R. Kunz, Program Manager

Approved by W. J. Lueckel — W. J. Lueckel, Chief Space Power Systems

Pratt & Whitney Aircraft DIVISION OF UNITED AIRCRAFT CORPORATION

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FOREWORD

This report was prepared by Pratt & Whitney Aircraft Division of United Aircraft Corporation, East Hartford, Connecticut, to describe the work conducted during the period June 1, 1963 to December 1, 1963 in fulfillment of Task I of Contract NAS3-2335, with Amendments 1 and 2, Experimental Investigation of Transients in Simulated Space Rankine Powerplants. This task consists of an investigation of methods of controlling the start and operation of a two-loop nuclear Rankine-cycle powerplant in space. The two-loop powerplant is characterized by a direct-condensing radiator. Investigations were conducted with a facility designed to simulate many aspects of the powerplant.

In order to limit this report to the presentation of new data, references will be made to data presented in an earlier report* under this contract. A detailed description of the experimental facilities and a summary of test results previously obtained may also be found in that report.

The experimental work reported here was performed during the months of July, October and November, 1963. The remainder of the report period was devoted to 1) installing an improved radiator simulator, (see Appendix A), and 2) changing the facility to simulate a three-loop powerplant configuration.

These tests concluded investigations on a two-loop powerplant configuration. Future testing will be conducted under Task II of the amended contract, with the powerplant simulator modified to three-loop configuration by the addition of an intermediate compact condenser and a liquid radiator loop.

*Hooper, J.R., and H.R. Kunz, Experimental Investigation of Heat Rejection in Nuclear Space Powerplants, Report PWA-2227, Volume 3, Space Powerplant Startup and Transient Operation Simulation, Report Period June 1, 1962 through May 31, 1963

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I. INTRODUCTION

Experimental investigations have been conducted on a facility designed to simulate many characteristics of a nuclear Rankine-cycle space powerplant. The purpose of the program is to locate possible problem areas associated with methods that have been proposed for starting and controlling such a powerplant in space, and to investigate possible solutions to these problems.

During the period covered by this report, investigations were conducted on a simulated two-loop powerplant configuration with a direct-condensing radiator. A schematic diagram of the powerplant simulator is shown in Figure 1. The components of the simulator were designed to use water as the working fluid in order to add greater flexibility to the experimental investigation. However, the components were also designed to approximate the characteristics of the components of a liquid-metal space powerplant with respect to such properties as fluid velocity and heat capacity. A detailed description of these simulator components may be found in Volume 3 of Report PWA-2227, pages 5-13. A redesigned radiator simulator was installed in the rig at the beginning of the period covered by this report. This design eliminated the thermal stress failures encountered with the original design. A description of the redesigned radiator is given in Appendix A. The rig was instrumented to obtain a continuous oscillograph recording of the fluid temperatures and pressures at the inlet and exit of each component during transients. Flow rates, electrical power input, fluid inventory, and several other variables were also recorded. A line drawing of the simulator, showing the location of the instrumentation and line sizes, is presented in Figure 2. The power loop is drawn to scale in this figure.

The approach taken in the test program was first to make exploratory investigations of different starting techniques on the simulator for the purpose of locating possible problem areas. Although the magnitude of the problem may not be accurately indicated for an actual powerplant of a specific design, the simulator was instrumented to obtain sufficiently detailed transient data so that the response of the system could be related to the physical constants of the system. Consequently, a basis for evaluating the data in terms of a given design is provided. Followup investigations were carried to the point where insight could be gained into the mechanisms underlying the problem, in order to provide data for possible future analytical treatment, and to suggest means of eliminating the problem. The program was intended to draw the attention of the powerplant designer to certain problems which might not otherwise become apparent until system testing was undertaken, and to indicate possible solutions to these problems.

This report presents data from a series of hard-fill starting tests, and discusses these test results. The hard-fill is one technique for starting a powerplant in space with the Rankine-cycle power loop initially evacuated. The soft-fill, which is another technique for starting with an initially evacuated power loop, was discussed in Volume 3 of Report PWA-2227. The weight savings which might result from use of the hard-fill starting method appear attractive, if it can be made workable. However, certain problems seem inherent in its use in a two-loop powerplant, and it was these problems which were the subject of the investigations described in this report.

II. SUMMARY

Experiments were conducted to investigate the feasibility of using a hard-fill method of starting a direct-condensing nuclear Rankine-cycle powerplant in space. In a hard-fill start, only the amount of working fluid required at steady state is injected into the evacuated Rankine-cycle power loop, and no accumulator is provided. The investigations were performed with a powerplant simulator using water and designed to simulate many of the fluid flow and heat transfer characteristics of such a powerplant.

Two problems appeared to arise because of this starting technique: 1) a small error in the amount of fluid injected may cause a significant change in the steady-state operating conditions of the powerplant, and 2) the required net-positive-suction head at the power-loop pump inlet probably cannot be maintained during the starting period unless additional control measures are added to the starting procedure. Three means of maintaining the required NPSH were investigated on the simulator. The most practical seemed to be to overfill the system during the starting period from a fill tank and a small pressure-regulating accumulator. The excess fluid is returned to the accumulator as the system approaches steady state.

Exploratory tests on a starting method in which the Rankine-cycle loop is filled with liquid prior to starting indicated that special care is required in providing means for the removal of excess fluid from the power loop, and in regulating the rate of heat addition and the flow rates during the starting period.

III. TWO-LOOP STARTING PROCEDURES

The general starting procedure used in the two-loop hard-fill tests described in this report was established earlier for a series of soft-fill tests in a simplified system, reported in Volume 3 of Report PWA-2227, pages 25-28. The description of this starting method which follows, holds for both the soft-fill and hard-fill procedures as defined later in this section. The description employs the notation used in Figures 1 and 2.

Before the starting procedure is initiated the power loop is evacuated. The temperature and flow rate in the sink for the radiator-simulator are set at their design values, and automatically maintained at these values throughout the test. Flow is established in the heater loop and the temperature of the heater loop fluid at the heater exit is brought to its design value, where it is maintained by regulating the power input to the heater.

The starting process begins with the opening of valve AV1, in the line from the fill tank to the power loop. Regulated pressure on the fluid in the fill tank forces fluid into the power loop at a rate controlled by the setting of valve MV3 (when the loop is filled at the pump inlet). In general the initial flow entering the boiler is held to a fraction of the design flow rate to avoid thermally shocking the heater. The closed radiator return valve AV6 prevents flow from entering the radiator and directs it into the boiler, where it is vaporized, and through the turbine simulator valve MV1. The vapor is condensed upon entering the radiator, and the radiator exit lines and manifolds begin to fill with liquid. The power-loop pump is started at some fixed time after the initiation of the start. Nevertheless, the pressure at the radiator outlet remains near the saturation pressure at the condensing temperature until the radiator tubes are partially filled with water. Only after the condensing heat transfer area in the radiator has been reduced to the value required to just condense the fluid at the incoming flow rate and the local condenser wall temperature does the pressure begin to rise. Opening the radiator return valve establishes circulation throughout the loop. A steady state is reached after the thermal lag in various parts of the system has been overcome.

The control system used to govern this starting procedure is one which is nearly minimal both in the number and the complexity of the controls involved. All valves have either fixed settings or on-off controls. It should be noted that certain auxiliary steps which might have to be taken in a space powerplant, such as providing lubrication for bearings and auxiliary power to start the pumps, were not considered.

The terms hard fill and soft fill are used in this report to designate two different means of accomplishing the starting procedure just outlined. They are defined as follows:

Hard Fill - The working fluid is injected into the power loop from the fill tank at a fixed predetermined rate which is independent of the driving pressure in the fill tank and the system pressure at the point of injection. That is, flow controls operate to maintain a preset flow rate from the fill tank during the fill period, and the fill tank pressure must simply be sufficient to maintain this flow rate. The exact amount of fluid required by the loop to produce the design operating conditions at steady state is injected during the fill period. When the correct amount of fluid has been injected, the fill tank is shut off from the loop and serves no further purpose.

Soft Fill - The working fluid is injected into the power loop from a fill tank in which the driving pressure is maintained at the design steady-state loop pressure at the injection point. Consequently, the fill rate is dependent on the difference between the driving pressure and loop pressure, and varies in response to changes in loop pressure during the starting period. The fill tank remains attached until a steady-state condition is attained, and an interchange of fluid between the fill tank and the loop may occur at any time. Typically, considerably more fluid is injected than is required at steady state due to the heat capacity of the radiator, which causes the condensing length to be much shorter during the initial period than after a steady-state heat rejection rate is reached. This excess fluid is returned to the tank as the system approaches steady state. The fill tank may also remain in the system throughout its operational life and serve as an accumulator. This allows the loop inventory to adjust to a change in operating conditions, while the pressure at some point such as the pump inlet remains constant.

Considering the implications of these definitions, certain advantages and disadvantages of the hard-fill starting method may be hypothesized. These served as a guide in formulating the test program on the simulator.

In particular, testing was set up to confirm the disadvantages postulated in the list below, and to look for means of alleviating these difficulties.

The hard-fill starting method provides close control of the flow rate during the starting period and thereby enables the powerplant to start without exceeding the possible dynamic control limitations of a nuclear reactor power source.

Use of the hard-fill starting method might effect weight savings in the design of a powerplant, as compared to the soft-fill method:

1) No pressure regulating mechanism is required for the fill tank

2) The fill tank need not be protected against meteoroid damage after the initial starting period.

3) Only the amount of working fluid required for steady-state operation need be carried aloft.

The disadvantages of the hard-fill method appear to consist of a loss in reliability in starting and operating the powerplant (although the pressure regulating mechanism for an accumulator in the soft-fill method may present problems in this respect). These disadvantages are as follows:

1) An error in the amount of fluid injected during a hard-fill start might cause a change in the steady-state operating conditions of the powerplant. One possibility of such an error exists because the power-loop inventory requirements must be established in ground testing, where stratification due to gravity may occur in two-phase fluid passages.

2) In the hard-fill start the power-loop pump inlet pressure will follow the radiator exit pressure from the time the fill-tank is shut off from the loop, while in the soft-fill start direct control over the pump inlet pressure may be maintained at all times during the start. The closed-loop radiator exit pressure will depend on the condensing rate in the radiator at a given time. In general, this pressure will be lower than the design value during the starting period unless heat is added to the radiators from a source external to the power loop.

These low pressures occur for the same reasons that caused over-filling in the soft-fill method. Should the pump inlet pressure fail to meet the minimum net-positive-suction head requirements of the pump, the result may be cavitation damage to the pump, or even loss of circulation in the loop. In a closed loop, loss of circulation results in a failure to start, since the pump inlet pressure can only fall further as the radiator cools.

3) If there is small loss of inventory from the system during its operation or a change in the operating characteristics of one of the components, then with fixed controls the output of the powerplant may change significantly unless a correction is made in the power-loop inventory. When the hard-fill method is used, no accumulator is available to make such inventory adjustments.

Two remarks should be made here concerning the disadvantages of the hard-fill method postulated above. First, they apply in all respects only to a two-loop powerplant. In a three-loop configuration, which uses an intermediate indirect condenser, it may be possible to both compensate for a small inventory error and to maintain control over the pump inlet pressure during starting, by controlling the flow of the radiator loop fluid through the intermediate condenser. Second, the hard-fill and soft-fill methods are not mutually exclusive. For instance, a hard-fill start may be conducted with an auxiliary accumulator in the system at the pump inlet which is added at the time the fill tank is shut off. The accumulator would regulate the pump inlet pressure and allow an adjustment of the loop inventory. Since this auxiliary accumulator could be considerably smaller than one required to hold the entire inventory, a weight saving might result.

V. TEST RESULTS

A. Steady-State Tests

Before starting tests were begun, a number of steady-state runs were made to obtain data on the nature and magnitude of the effects resulting from a measured change in the power-loop inventory. Since the disadvantages hypothesized for the hard-fill starting method result from operating the power loop without an accumulator, it is necessary to know the operating characteristics of the simulator with a fixed power-loop inventory, in order to be able to evaluate the transient data.

For the purpose of these tests, an arbitrary standard steady-state operating point was first established with an accumulator attached at the power-loop pump inlet. At this standard point about 10°F of superheat was obtained at the turbine simulator inlet. The flow rate was set lower than for the starting tests so that variations around this point could be investigated without exceeding the power limitations of the rig, but other conditions were similar to the design operating conditions used in the starting tests.

Once the standard operating point was established, the accumulator was valved off from the power loop, thereby fixing the standard inventory in the loop. All controllable powerplant parameters were then held constant with the exception of the power loop flow rate which was treated as an independent variable. The fixed quantities included the temperature and flow rate in the heater loop and sink loop and the setting of the turbine simulator valve. The power-loop flow rate was changed with a throttling valve, located downstream of the pump. Since the pump was operated at a constant speed, the boiler pressure changed as the setting of the throttling valve was varied.

With this standard inventory in the power loop, steady-state data was obtained at several settings of the pump throttling valve. Next, a measured quantity of fluid was injected into the power loop, and again data was obtained at several power-loop flow rates and corresponding boiler pressures. This procedure was repeated to obtain data at four different inventories. The data is plotted in Figures 3 and 4.

In order to provide a basis for comparison with other systems, the change in inventory has been expressed as a percentage of the amount of liquid that would fill the radiator-condenser. A comparison on this basis is more significant than one where the change is expressed as a percentage of the total loop inventory, since the volume of the liquid portion of the loop may vary from system to system. The liquid portion of the power loop in the simulator probably has a greater volume than a space system would have, because of the large liquid volume of the pumps used.

The turbine simulator valve was choked at all times during this test, and consequently provided a relationship between upstream conditions and flow comparable to that of a choked turbine. The fact that no enthalpy was removed by the turbine simulator caused the conditions of the flow entering the radiator-condenser to differ from those that would result with a turbine in the system. However, since the volume in the condenser required to take out the enthalpy removed by a turbine is comparatively small, the difference in condenser inventory for the two cases should not be great.

In Figure 3 boiler exit pressure is plotted against flow rate at different loop inventories. Since the turbine simulator valve was choked, the flow rate through the valve was dependent only on the pressure and temperature upstream of the valve when the fluid at that point was superheated, or on the pressure and quality when it was saturated. The temperature or quality at the valve inlet was determined by the boiler characteristics for a given boiler flow, exit pressure, and inlet temperature. Since the variation of boiler inlet temperature can be shown to be unimportant in the range of data obtained, the boiler and choked valve characteristics provide two functional relationships from which the boiler exit temperature or quality can be eliminated, and the boiler exit pressure is seen to be uniquely related to flow. The boiler exit superheat or quality, and the temperature of the fluid entering the boiler are shown in the figure next to each data point.

Adjustment within the system to a change in inventory was achieved by a change in the condensing pressure such that two conditions were fulfilled:

1) the heat accepted by the boiler equalled the heat rejected by the radiator-condenser, and

2) the available fluid filled the loop at the local equilibrium temperatures and pressures.

Since the per cent by weight of vaporized fluid in the system is very small, the effect of change in vapor inventory on the inventory-condenser pressure relationship is small. Consequently a change in the locations of the liquid interfaces in the boiler and condenser (visualizing a discrete location for these interfaces) is the principal means by which the above two conditions are satisfied when the system inventory is changed. At a constant power-loop flow rate, the interface locations will change primarily due to a change in the heat transfer rates in the boiler and condenser. If the heater loop temperature and the sink temperature remain constant, then the heat transfer rates will depend on the pressure of the power-loop fluid in the boiler and condenser. This dependency arises in two ways, 1) the heat transfer coefficient of the power loop fluid in the boiler and condenser is pressure dependent, since liquid and vapor properties of steam at saturated conditions are a function of pressure, and 2) the temperature difference between the heater loop and the power loop fluid in the boiler, and between the power loop fluid and the sink in the radiator-condenser, depend on the saturation temperatures of the power-loop fluid at the pressures existing in the boiler and condenser. The latter is the more important effect.

The pressure dependencies of heat transfer in the boiler and condenser, then, primarily determine the form of the inventory-condenser pressure relationships. An interconnection between the boiler and condenser is provided by the fact that the fluid enthalpy must be nearly equal at the boiler exit and the condenser inlet, and at the condenser exit and boiler inlet. Another interconnection is provided by the fact that the sum of the pressure drops around the loop must equal zero.

Figure 4 shows condenser pressure data plotted against flow rate at various fixed inventories. Radiator outlet temperature data was crossplotted and lines of constant outlet temperature are located on this map.

It can be seen in Figure 4 that as the closed-loop inventory increases at a fixed flow rate, the condensing pressure increases. This increases the temperature difference between the condensing fluid and the sink and decreases the condensing length so that there is more liquid in the radiator. At the same time the temperature of the fluid at the radiator outlet decreases and the fluid enters the boiler with more subcooling so that the inventory in the boiler may also increase.

A line of 100 per cent quality at the boiler exit has been plotted in Figure 4. If it is undesirable to allow liquid droplets to enter the turbine, then this line represents one limit to the allowable variation in operating conditions. Another limit is the onset of pump cavitation. An extrapolated line showing the conditions at which saturated liquid would be present at the pump inlet is plotted in Figure 4. Pump cavitation will occur at some condenser pressure above this line. The location of the cavitation line will depend on the characteristics of the pump or pumps used.

When the turbine simulator becomes unchoked, the form of the relationships plotted may be expected to change, since the flow rate becomes dependent on the condenser pressure as well as the boiler exit pressure. The line at which this unchoking occurs is shown in Figure 4.

Probably the most important variable affected by an error in inventory would be the power output of the turbine. The variation with inventory of the power output of a turbine may be approximately calculated for the conditions of this test by using the turbine inlet temperatures, inlet and outlet pressures, and flow rates given by the data at different inventories. The results of such calculations are plotted in Figure 5. It was arbitrarily assumed that the flow was varied so as to maintain 10°F of superheat at the turbine inlet as the inventory was changed. Because of the approximations involved in the calculations, this curve should be considered valid only as indicating a trend. However, the rate of decrease of turbine power output with an increase in inventory is shown to be significant.

B. Hard-Fill Starting Tests

Starting tests were made to investigate problems with the hard-fill starting method in a two-loop system. Since difficulty in maintaining the required net-positive-suction head at the power-loop pump inlet during the starting period could be foreseen, the tests were set up with special attention to variables that might affect the pump inlet pressure.

1. Hard-Fill Starting Test Program Approach

The method of power-loop flow control simulated during the starting period in these tests was one in which the flow is automatically held at the desired value by a valve downstream of the pump (and downstream of the line from the fill tank when the loop is filled at the pump exit). The location of this valve in the system would be the same as MV2 in Figure 2. In this mode of start the fill tank provides the driving pressure upstream of the control valve during the fill period, and pump operation is not required to maintain the fill rate. The boiler inlet pressure during the fill period depends on the fill tank pressure and (since the valve opening adjusts to maintain the desired flow) on the fill rate selected.

In these tests, a starting procedure was simulated in which the pressure on the fill tank is maintained at the design pump outlet pressure, and in which the fill rate is maintained at 30 per cent of the power-loop design flow rate. After filling, the power-loop flow rate is stepped up to its design value by opening the radiator return valve. The pump and preset throttling valve then allow design flow to circulate through the system. A low fill rate is desirable to prevent possible thermal shock to components of the power system, and to prevent the control response characteristics of the nuclear reactor from being exceeded. The 30 per cent figure was selected as the lowest feasible fill rate for hard-fill tests after a consideration of data from the previous soft-fill test series. In the soft-fill tests, overfilling of the loop was extreme below a 30 per cent fill rate, indicating that in a comparable hard-fill test the pump would have to operate under extreme cavitation conditions.

In the absence of an automated valve to control the flow, the following procedure was employed to simulate a starting process in which the power-loop flow is automatically held constant at 30 per cent of the design flow rate by the pump throttling valve during the fill period, with a step up to design flow when the fill tank is shut off. The power-loop-pump throttling valve was preset to give the standard flow rate at steady-state conditions. The fill rate was controlled by a fixed valve in the line from the fill tank. The pump was started soon after fluid was available at its inlet (about 20 seconds after injection was initiated, except during the one test in which loop was filled at the pump exit). Since the pump lowered its inlet pressure to a very low value, the pressure drop across the fill control valve was nearly constant, causing the fill rate to be almost constant at the desired value. Also, the boiler inlet pressure was similar to that which would occur were an automated valve controlling the flow from a fill tank held at the normal pump exit pressure.

This method of operation was possible because of the operating and cavitation characteristics of the power-loop pump used in the simulator. A water pump having a reasonable efficiency at the simulator flow rates had to be used, and two turbine pumps in series were selected as the best standard item available. However, the required NPSH of these turbine pumps is less than one psi, much lower than the required NPSH of a typical centrifugal pump for use in a liquid-metal space powerplant. Also, the sharp break in output at the cavitation point that is typical of a centrifugal pump does not occur. Nevertheless, as noted above, the conditions in the simulator, downstream of the pump throttling valve during the fill period when the loop was filled at the pump inlet, were entirely analogous to those in a powerplant in which the pressure on the fill tank is constant at the steady-state pump exit pressure, and the fill rate is automatically held constant by the pump throttling valve. The one exception to this statement is the occasional pressure fluctuations caused in the simulator by starving the pump inlet during the fill period.

There is, however, another and more important consequence of the cavitation characteristics of the simulator pump. After the fill tank is shut off in the hard-fill starting method, the pump must maintain circulation with its inlet near the radiator exit pressure. This is one major problem in the hard-fill starting method. The simulator pump continued to operate under conditions that would obviously cause a centrifugal pump to cavitate. Therefore it was necessary to establish an arbitrary criterion for the minimum allowable pump NPSH for a successful start. This criterion could be applied to the pump inlet pressure data recorded during the tests.

The designer of a powerplant must make a trade-off between the margin of safety provided by a large NPSH at the pump inlet at design conditions, and the improved powerplant performance resulting from a lower NPSH. Although this trade-off may be different in different powerplants, for the purpose of these tests, the minimum allowable NPSH was set at about 60 per cent of the NPSH available at design conditions. This resulted in designating a minimum allowable NPSH of 16 psia, about 10 psi below the NPSH available at standard steady-state conditions. With this arbitrary reference the relative effect of the different powerplant variables could be compared.

2. Hard-Fill Starting Test Program Outline

Using the general starting procedure outlined in Section III above, hard-fill starts were made. During the fill period the power-loop inventory was visually monitored with the apparatus shown schematically in Figure 2. When the correct inventory (as established by a previous steady-state run) had been injected, the valve in the fill line was closed.

The standard steady-state operating conditions towards which the starting process was aimed were made similar to those used in many of the soft-fill tests reported in Volume 3 of Report PWA-2227 in order to facilitate comparisons between the two sets of data:

power input to heater - 100 KW
power-loop flow rate - 325 lb/hr
boiler shell flow rate - 32,000 lb/hr
boiler inlet pressure - 155 psia
boiler exit pressure - 140 psia
boiler exit superheat - approximately 10°F
boiler shell inlet temperature - 415°F
boiler shell exit temperature - 405°F
radiator exit pressure - 30 psia
sink temperature - 80°F
radiator exit temperature - 150°F

Two of the three available radiator-simulator segments were used. Usually the liquid interface was located about two-thirds of the distance along the radiator tubes from the inlet at steady state.

The time plots of the more important variables recorded during the hard-fill starting tests are presented in Figures 6 through 16. It should be noted that the radiator exit pressure P10 is equivalent to the pump inlet pressure after the radiator return valve has been opened. Before this valve is opened, the pump inlet pressure is unimportant since pump operation at this time is not a necessary feature of the hard-fill starting method. Each of the tests for which data is presented is listed below.

3. Description of Tests

The hard-fill starting tests which yielded a complete transient record of the significant variables are discussed below. A list of those tests follows:

Tests Nos. 1 and 2 (Figure 6) - Standard hard fill
Test No. 3 (Figure 7) - Increased radiator volume
Test No. 4 (Figures 8 and 9) - Decreased power-loop flow rate
Test No. 5 (Figure 10) - Intentional underfill
Test No. 6 (Figure 11) - Fill at pump exit
Test No. 7 (Figures 12 and 13) - Segmented radiator (semi-hard-fill)
Test No. 8 (Figure 14) - Preheated radiator
Test No. 9 (Figure 15) - Overfill and ejection to an accumulator, (semi-hard-fill)

Tests No. 1 and 2 (Figure 6) - Standard Hard Fill

Two hard-fill starting tests were made which led to the standard steady-state operating conditions.

In the first test, the radiator return valve was opened by a pressure switch in the radiator return line set at a pressure slightly below the design steady-state value. However, the pressure switch was not activated by the time the correct inventory had been injected and the fill tank shut off. Although fluid was no longer being supplied to the pump inlet after the fill tank was shut off, the system pressures were maintained, probably by expansion of vapor in the boiler. About 10 seconds elapsed after the fill tank was shut off before the small additional pressure rise necessary to open the radiator return valve occurred.

Since a centrifugal pump probably would not be able to maintain pressure under similar conditions, the controls were changed so that the radiator return valve would be automatically opened when the fill tank was shut off, and a second test was made. Data from the two tests indicated an essentially similar system response during the starting period. That is, the data plotted in Figure 6 describes both tests, with the exception of the 10-second period between the closing of the fill valve and the opening of the radiator return valve in the first test. Controls which opened the radiator return valve when the fill valve was closed were used in all succeeding tests.

During the standard start, the pump inlet pressure fell to approximately 10 psia at its minimum point after the fill tank was shut off. This is 6 psi below the limit established for the allowable NPSH, so that this starting procedure must be regarded as inadequate. A plot of the NPSH at the pump inlet during the time immediately following the closing of the fill valve is shown in Figure 16.

Test No. 3 (Figure 7) - Increased Radiator Volume

A test was made to investigate the effect of using a radiator with a larger volume in its fluid passages. For this test an additional radiator-simulator segment was added in parallel to the two already in use. The sink conditions were adjusted to keep the standard heat rejection rate, and the increased inventory required for standard conditions at steady state was injected during the start. The time plot of the pump inlet pressure after the fill tank was shut off showed no significant difference from the standard test.

Test No. 4 (Figures 8 and 9) - Decreased Power-Loop Flow Rate

This test differed from the standard in that the steady-state power-loop flow rate was reduced from 325 to 240 lbs/hr, and the heater-loop temperature was reduced from 415 to 405°F to keep the liquid interface in the boiler at about the same place at the lower flow. There was a slight unintentional underfilling of the loop by 100 milliliters (approximately 2 per cent of the total standard inventory). When allowance has been made for this it appears that the pump inlet pressure would not have been significantly lower than during the standard start. This test, together with Test No. 9, indicates that an increase in the system volume relative to the flow rate does not in itself adversely affect the pump inlet pressure during the starting process.

No importance should be attached to the fluctuations in boiler pressure indicated in Figure 8, during the period before the radiator return valve was opened. These fluctuations occurred because the pump inlet was starved, as noted before, and would not occur were the fill rate controlled downstream of the pump.

Temperatures were recorded during this test from thermocouples brazed on the outside of one of the radiator tubes. The temperature plots at three of these locations are shown in Figure 9. As the temperature at each thermocouple approached the saturation temperature a regular temperature fluctuation occurred, which evidently indicated the fluctuation of the liquid interface beneath the thermocouple. This means of locating the position of the interface, which agreed well with evidence from surrounding temperatures, was used to prepare the plot of interface location in Figure 9. The history of the interface location was similar in most other tests.

Test No. 5 (Figure 10) - Intentional Underfill

A test was made in which the fill tank was shut off from the power loop when the amount of fluid injected was less than the standard inventory by about 500 milliliters. This is about 10 per cent of the total standard inventory, or about 37 per cent of the amount that would fill the radiator. As was to be expected, the pump inlet pressure was extremely low just after the radiator return valve was opened, and the NPSH was below the minimum allowable at steady state.

Test No. 6 (Figure 11) - Fill at Pump Exit

In this test, the power loop was filled at the pump exit instead of the inlet. The pump was started a few seconds after the fill tank was shut off and the radiator return valve opened. The pressure drop across the valve in the fill line was sufficient to prevent the unstable boiler flow which had been found to occur with a soft accumulator (Volume 3 of Report PWA-2227, pages 21-24). Filling at the pump exit in this type of hard-fill start appears to present no special problems as compared to filling at the pump inlet, although the pump inlet pressure was slightly lower after the radiator return valve was opened. At the same time, the possible advantage of filling at the pump exit which was postulated for the soft-fill start in PWA-2227 may not apply here. For soft-fill starts the fill tank pressure is at the steady-state value desired for the system at the point of injection. When the injection is at the pump inlet, the fill tank pressure may be too low to provide sufficient pressure at the turbine inlet to start the pump. However, when injection is at the pump exit, the fill pressure would be high enough to provide sufficient pressure at the turbine inlet to start the pump. In the first case, the pump might have difficulty in starting with a boot-strap operation and auxiliary power may be required. In hard-fill starts with injection at the pump inlet, the fill tank pressure may be high enough to provide sufficient pressure at the turbine inlet to start the pump.

The starting procedure in all of the tests previously described proved inadequate since the minimum requirements established for the available NPSH at the pump inlet were not met. The following three tests were specifically directed towards investigating likely means of preventing pump cavitation during the starting period. Two of these tests were hybrid in nature, and did not conform strictly to the hard-fill definition.

Test No. 7 (Figures 12 and 13) - Segmented Radiator (Semi-Hard-Fill)

A starting procedure that might be applicable to a powerplant with a segmented radiator was attempted. The fill tank was located at the pump inlet, and the pressure in the fill tank was regulated at the standard pump inlet pressure. The radiator return valve was opened by a pressure switch set at 25 psia which operated from radiator exit pressure. One of the two radiator simulator segments was open to flow from the beginning, but the inlet of the other segment was opened only after the heat capacity of the first segment had been overcome and the fluid temperature at the radiator outlet rose to 150°F. The only element of the hard-fill start remaining in this test was that the fill tank was cut off after the steady-state inventory had been injected.

The outlet valve of the second radiator segment was initially closed, and was opened by a pressure switch within the segment set for 10 psia. Previously a test had been made under identical conditions but with the pressure switch set for 25 psia. In the earlier test, the pressure failed to activate the switch by the time the fill tank was cut out. As a result, the second segment simply filled with liquid, reducing the pump inlet pressure until the pump cavitated.

In the successful test, shown in Figure 12, it can be seen that the time required to inject the standard inventory was prolonged by the delay in opening the second segment, together with the dependency of the fill rate on the pressure drop across the fill valve. At one point there was actually back flow into the fill tank. Since through-flow in the power loop had been established early by the opening of the radiator return valve, much of the heat capacity of the radiator had been overcome by the time the fill tank was cut off. The time plot of the interface location in the radiator is shown in Figure 13. The data was obtained from the middle tube of the initially open segment. It appears that in this tube at least, the interface was pushed to the end of the tube just before the second segment was opened. This starting procedure was successful in providing an adequate pump inlet pressure at the time the fill tank was cut off. However, it was also extremely complicated, and from the point of view of reliability, it is desirable to look for a simpler alternative.

Test No. 8 (Figure 14) - Preheated Radiator

Since the major part of the thermal lag during the starting process originates in the radiator, it is obvious that steady-state conditions (and in particular, the steady-state pump inlet pressure) would be attained more quickly if the radiators were preheated. In this otherwise standard hard-fill test, the radiator-simulator was heated to a temperature of 140°F before the starting process was initiated. This was approximately 10°F below the steady-state radiator fluid outlet temperature, and 100°F below the condensing temperature at the steady-state condensing pressure. The sink was maintained at 140°F throughout the starting period, which is equivalent to continually adding enough heat to a radiator to maintain the initial temperature at the outside surface during the starting period.

Figure 16 shows the available NPSH to be marginal in terms of the arbitrary criterion established for a successful start. It was just below the lower limit of 16 psi at the time the fill tank was cut out, but quickly rose to an acceptable value. Since the initial radiator temperature could have been set higher, it is probable that a successful start can be made by preheating the radiator. However, the equipment required to preheat a radiator and to maintain control over its temperature might be heavy enough to offset the advantages gained by using the hard-fill starting technique.

Test No. 9 (Figure 15) - Overfill and Ejection to An Accumulator, (Semi-Hard-Fill)

In this test an attempt was made to maintain the minimum allowable pump NPSH after the fill tank was shut off and the radiator return valve opened by injecting more fluid than was required at steady state. A rough estimate of the required excess inventory was made on the basis of steady-state data and previous starting tests. This additional inventory (approximately 250 milliliters) was injected during the fill period. As the system approached steady state, the excess fluid was ejected into an accumulator in which the pressure was regulated at the desired pump inlet pressure.

The plot of the available NPSH in Figure 16 shows that this procedure was marginally successful in terms of the arbitrary criterion established for the minimum allowable NPSH. However, the plot of the radiator exit pressure in Figure 15 indicates a possible limitation to the use of this procedure. The radiator exit pressure rose above 40 psia, more than 10 psi above the design value. Since the steady-state available NPSH might be designed closer to the minimum allowable NPSH, more fluid would have to be injected to maintain this minimum, and the pressure in the radiator could go very high before the return valve was opened, because, as can be seen in Figure 4, the rate of increase of radiator pressure with inventory increases rapidly at higher radiator pressures. Consequently, overstressing of the radiator structure might result.

If this means of maintaining the minimum NPSH during starting can be used without overpressurizing the radiator, then it would seem to be the least costly modification that would make the hard-fill method workable. Although the weight advantage of storing only the fluid required at steady state would be lost, the only additional control needed is a provision for ejecting the excess fluid, which could be simply dumped through a relief valve.

This procedure may be further modified to eliminate the danger of overpressurizing the radiator, by including a pressure-regulated accumulator in the system at the pump inlet. Only the steady-state inventory would be injected from the fill tank. When the fill tank is cut off and the radiator return value opened, a valve in the line to the accumulator would be opened. The excess fluid would then be supplied from the accumulator, which would maintain the radiator pressure and pump inlet pressure near their design values. The history of such a start would be nearly identical to those of the soft-fill starts reported in Volume 3 of Report PWA-2227, but an advantage over the soft-fill procedure may result because of the reduced size of the accumulator.

C. Other Tests

Some miscellaneous starting tests were made during the period covered by this report. Results of interest from these tests are described briefly below. Since these tests were somewhat outside the main objectives of the program, the tests were only exploratory, and no systematic followup investigations were made of the phenomena observed.

1. Liquid Filled Starts

Two starting tests were made in which the power loop was initially filled with liquid. In this method of starting, liquid must be removed from the power loop as vapor is generated in the boiler and fills the region between the boiler and condenser. An accumulator in which the pressure was maintained at the design pump inlet pressure was attached to the loop at the pump inlet to receive the excess liquid. No attempt was made to regulate the power-loop flow rate during the starting period in these tests;the valves were set as required for steady-state conditions. In this respect, and in certain others noted below, these tests were not designed to use the most favorable starting conditions, but to provide an indication of the sort of problems that might arise.

Starting Test No. 1 (Flow Initiated First)

The first test consisted of the following sequence of events. The sink loop temperature and flow rate were set at their design value. Flow was established in the heater loop and power loop with flow control valves at their design setting. The fluid in both these loops was initially at room temperature. To initiate the start, the heater power was turned on. Until the heater loop temperature approached the design value, the maximum power of approximately 120 KW was provided to the heater. This heat addition resulted in a temperature rise in the heater loop of about 55°F/minute.

Soon after vapor generation began in the power loop, pressure fluctuations centering around the radiator were detected. The fluctuations occurred at a rate of two or more per second and were of high amplitude. These continued, with varying amplitude, until after the heater-loop temperature had reached its design value, and then disappeared shortly thereafter. These fluctuations were sufficiently severe to threaten radiator structural failure and gaskets were blown out of one of the manifolds on the simulator. An oscillograph recording of a period during which system temperatures were approaching their design values, but before the fluctuations attenuated, is shown in Figure 17. The radiator pressure traces and their magnitudes are indicated. Although more extensive instrumentation of the radiator simulator would be required for complete verification, the available evidence strongly indicates that the mechanism which produced the pressure fluctuations was the sudden collapse of vapor bubbles or pockets in the radiator tubes, and possibly the manifolds as well. A sharp pinging sound which seemed to come from the center radiator segment accompanied the fluctuations. The data suggests that late in the starting period there was alternately vapor in the exit manifold and a back flow of liquid into the center segment.

The test indicated that in this mode of start, the liquid interface may be pushed toward the ends of radiator tubes nonuniformly by the constantly increasing volume of vapor in the system, with a buildup and sudden collapse of vapor pockets in the radiator passages. This tendency is undoubtedly accentuated by any maldistribution of flow in the radiator passages. Control over the power-loop flow rate and the rate of power input to the heater, as well as an optimum design of the accumulator receiving the excess liquid, might do much to alleviate this condition.

Starting Test No. 2 (Heating Initiated First)

The second test consisted of the following sequence of events. The sink loop temperature and flow rate were set at their design value. Flow was established in the heater loop. With the power-loop pump off, the heater was turned on and the heater loop brought to its design temperature. During this time, vapor filled the boiler tubes and forced a corresponding amount of liquid into the accumulator. To initiate the start, the power-loop pump was started.

In less than three seconds the radiator pressures rose to several times their design values, causing the rupture of a radiator manifold and the termination of the test. Figure 18 shows the oscillograph recording during this period, which is not clear since several pressures go out of range.

The radiator pressures rose due to the inability of the system to force liquid into the accumulator at a rate corresponding to the very rapid increase in the system vapor volume which occurred. A restriction in the line to the accumulator aggravated this condition. Control over the power-loop flow rates and a better accumulator design might alleviate the problem in this case also.

2. Dry Starts with Freezing Conditions in the Radiator

Two soft-fill starting tests were made with conditions similar to those reported in Volume 3 of Report PWA-2227, except that the sink temperature was set below the freezing temperature of the power-loop fluid. The radiator simulator tubes were allowed to come to the sink temperature before fluid was injected into the power loop. A light frost formed on tubes at this temperature. Consequently it was possible to tell whether or not fluid was freezing in the tubes, by the persistence or disappearance of the frost.

The first test was made with the sink temperature at 10°F, and the second at 0°F. In the first test, fluid in two or three tubes at the end of each radiator segment froze and remained frozen for a period of several minutes, while no freezing was apparent in the remainder of the tubes. The frozen tubes were eventually thawed by conduction from nearby tubes which were operating normally.

In the second test, fluid in one-half to two-thirds of the tubes of each segment initially froze. With the heat rejection capability of the radiator impaired by the loss of a considerable portion of its effective surface area, the fluid temperature at the radiator exit quickly rose and caused the pump to cavitate. It is apparent that in a marginal freezing situation the amount of flow maldistribution in the radiator tubes would be important in determining the success or failure of the start.

V. CONCLUSIONS

The following conclusions are drawn from the test results discussed in this report.

A. The principal problem in starting a two-loop powerplant with the hard-fill technique is to maintain the NPSH requirements at the power-loop pump inlet. Control measures in addition to those by means of which the hard-fill procedure was defined appear to be necessary. Once the fill tank is shut off from the power loop, the pump inlet pressure will follow the radiator exit pressure. Assuming little thermal lag in other parts of the system, the radiator exit pressure will primarily depend on the interrelated effects of 1) the temperature of the radiator metal at the end of the condensing region, and 2) the total loop inventory. With only the amount of fluid in the loop required at steady state, the radiator exit pressure will necessarily be below its steady-state value until the thermal lag in the radiator is overcome. Testing on the simulator indicated that in a powerplant in which the steady-state pump inlet pressure is designed to provide a reasonably small NPSH margin, the pump would cavitate during a standard hard-fill start.

B. Locating the fill tank at the pump outlet, as compared to locating it at the pump inlet, presents no special problems in a hard-fill start.

C. Modifications may be made in the hard-fill procedure to control the radiator temperature so as to assure an adequate pump inlet pressure during the starting period. The radiator temperature may be controlled by including a sufficient radiator warmup time in the starting procedure, as in the segmented-radiator test, or by heating the radiator with an external heat source. However, the complications in the control system and the weight of the additional equipment might negate any advantage in using a hard-fill starting procedure in a space powerplant.

D. The simplest modification that would make the hard-fill starting method workable seems to be an overfilling from the fill tank during the starting period. If enough fluid in excess of the amount required at steady state is injected, the minimum allowable NPSH can be maintained. Care must be taken that this does not cause pressure limitations in the radiator to be exceeded. The excess fluid could be dumped through a relief valve. However, using a constant pressure accumulator to supply and receive the excess fluid in this method would eliminate the danger of overpressurizing the radiator and would eliminate the possible problems noted in F. below.

E. Operation of the power loop without an accumulator can present problems since certain two-loop steady-state powerplant operating parameters are extremely sensitive to a small change in inventory. Problems may arise because 1) there is a possibility of injecting an incorrect amount of fluid during starting, 2) leaks may occur in the system during operation, and 3) operating characteristics of some components may change with time.

F. In a two-loop liquid-filled starting method, care must be taken in the procedure for removing excess liquid from the power loop, and in regulating the rate of heat addition and the flow rates during the starting period. When the heater-loop temperature is increased after circulation has been established in the power loop, there is a tendency for pressure fluctuations to occur in the radiator. When circulation is started after the heater loop has been brought to design temperature, there is danger of overpressurizing the radiator. Further work needs to be done to determine the seriousness of these problems in any particular system.

G. When the initial radiator temperature is below the freezing point of the power-loop fluid, plugging of some tubes can occur due to freeze-up, with a larger number of tubes becoming plugged at lower initial radiator temperatures. Pump cavitation can occur if enough tubes become plugged to result in a significant loss of radiator heat rejection capacity. Further work should be done in this area to explore the full extent of the problem.

APPENDIX A

Redesign of Radiator Simulator

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Redesign of Radiator Simulator

The radiator simulator originally installed in the rig (described in Volume 3 of PWA-2227) developed progressive structural failures as a result of the stresses caused by thermal cycling, particularly the thermal cycling due to continually starting and shutting down the power-plant simulator. As soon as these structural failures were detected, the radiator simulator was redesigned to withstand the thermal stresses that were encountered. The redesigned radiator was constructed and installed in the rig at the beginning of the contract period covered by this report and has operated satisfactorily with no failures due to thermal stresses.

The new radiator simulator was designed to have essentially the same overall heat capacity and the same geometry for the power-loop fluid flow passages as the original design, in order to assure a similar transient response. Three identical segments of 61 tubes each were used as in the original design (see Figure A1). The all-aluminum construction employed in the original design was replaced by stainless steel tubes carrying the power-loop fluid and by a copper slab which provides the major portion of the heat capacity. These materials were chosen because of their better high temperature strength and the ease with which the tubes could be brazed to the slab. The tubes were placed in grooves cut into the top of the slab, as shown in Figure A1, and then brazed. This prevented the bowing of the tubes encountered in the original design.

Heat is rejected from the radiator simulator by convection to a fluid flowing in a 1-1/2 inch high passage beneath the slabs. The sink loop can be operated with independent control over both the temperature of the sink fluid and the heat transfer coefficient between the bottom of the slab and the sink fluid, as in the original design. (The heat transfer coefficient is controlled by controlling the flow rate of the sink fluid). Consequently different combinations of initial radiator temperature and steady-state heat rejection rates may be employed during starting tests.

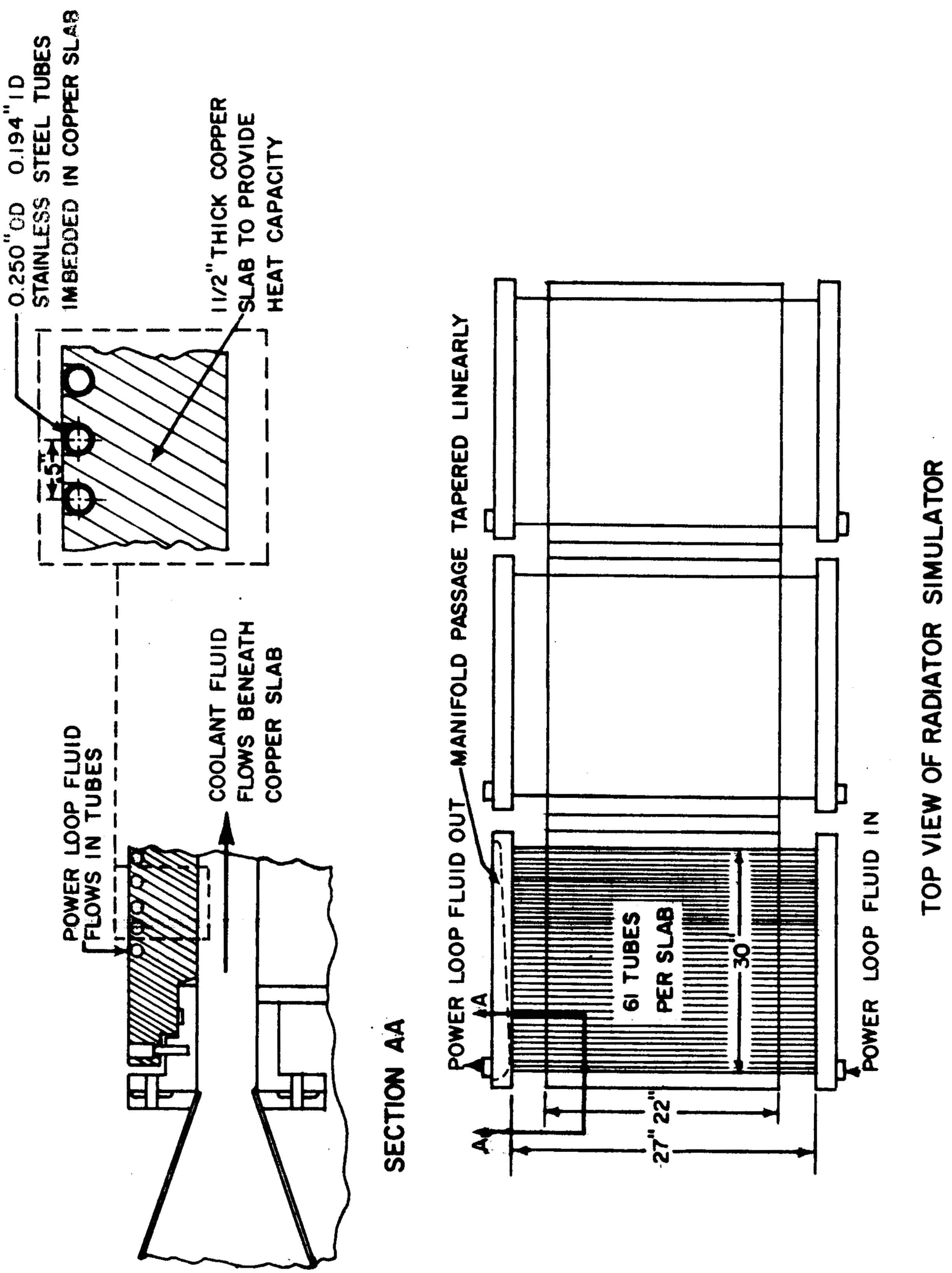

Figure A1 Radiator Simulator Redesign

APPENDIX B

Figures

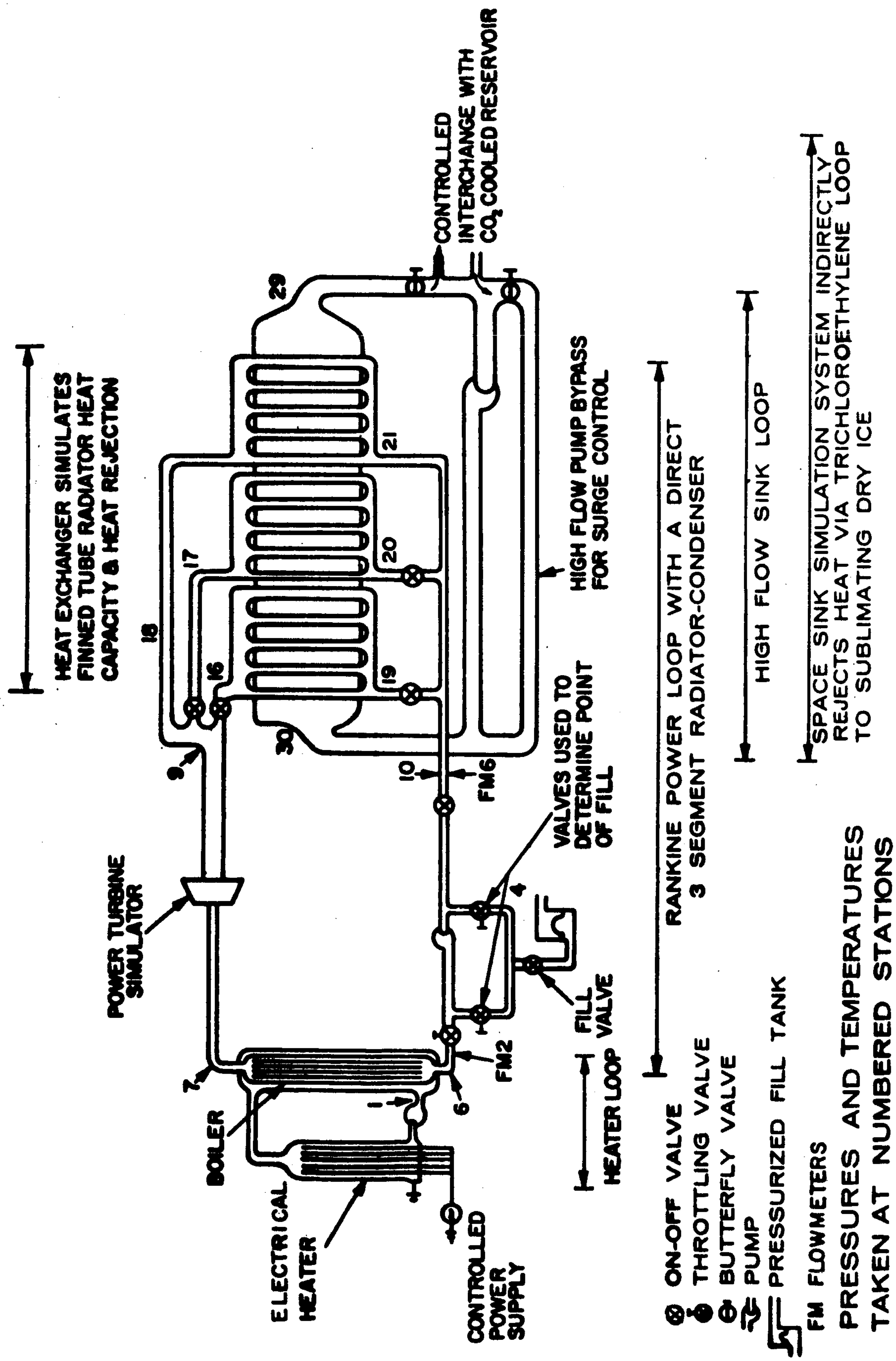

Figure 1 Schematic Diagram of Powerplant Simulator

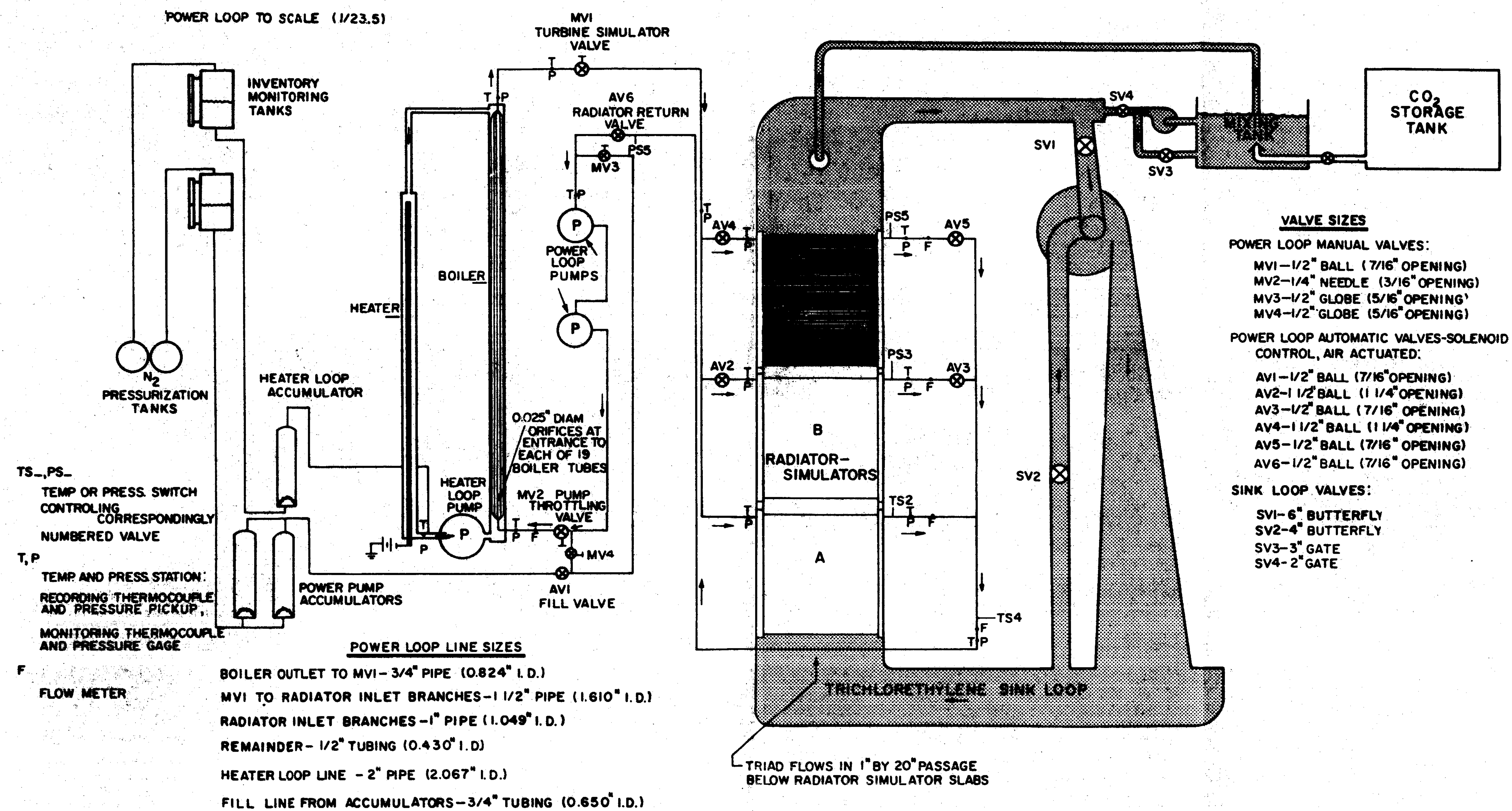

Figure 2 Powerplant Simulator Arrangement

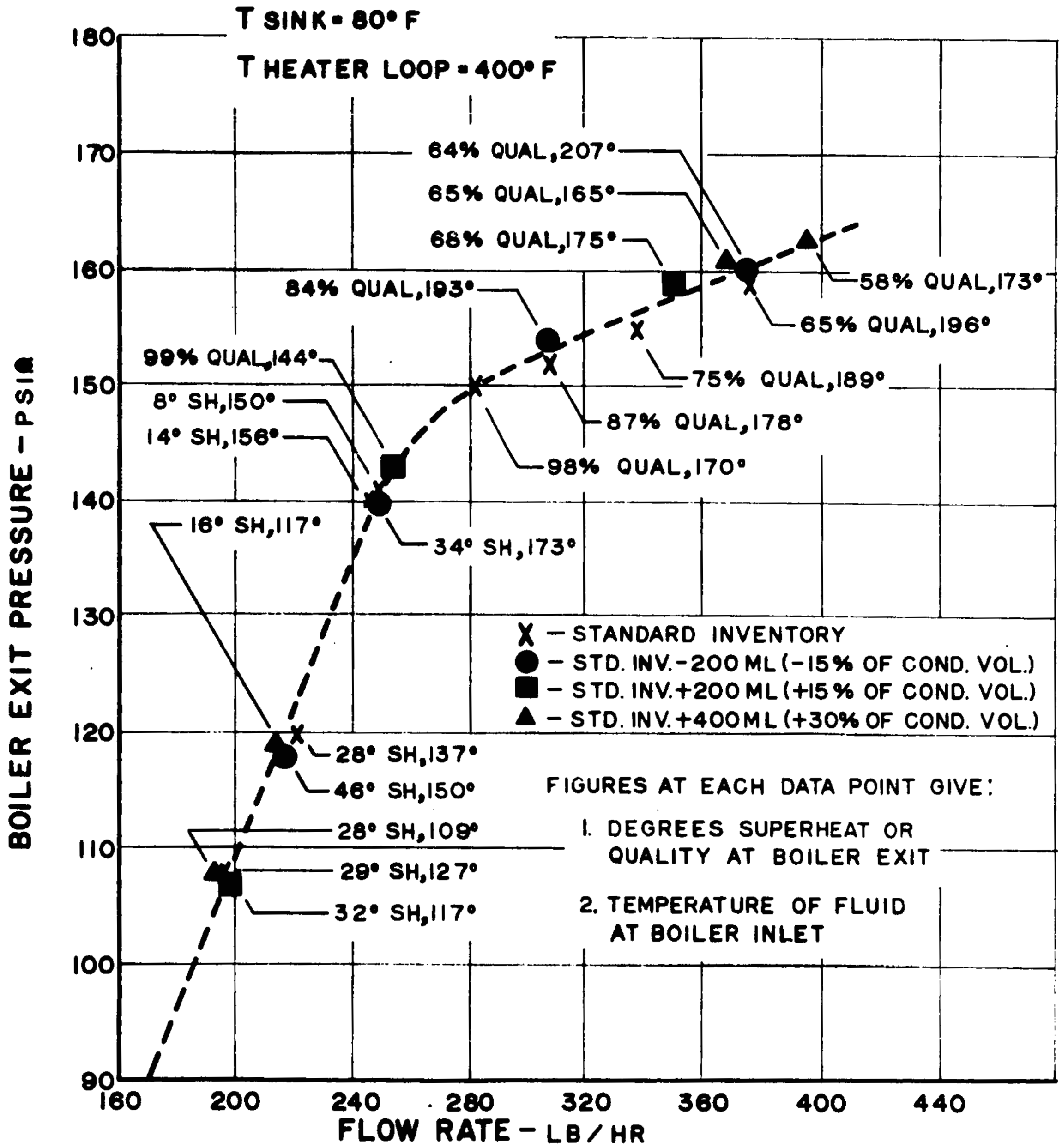

Figure 3 Steady-State Boiler Outlet Pressure vs Flow Rate at Different Fixed Total Loop Inventories. Constant Turbine Simulator Restriction (Choked)

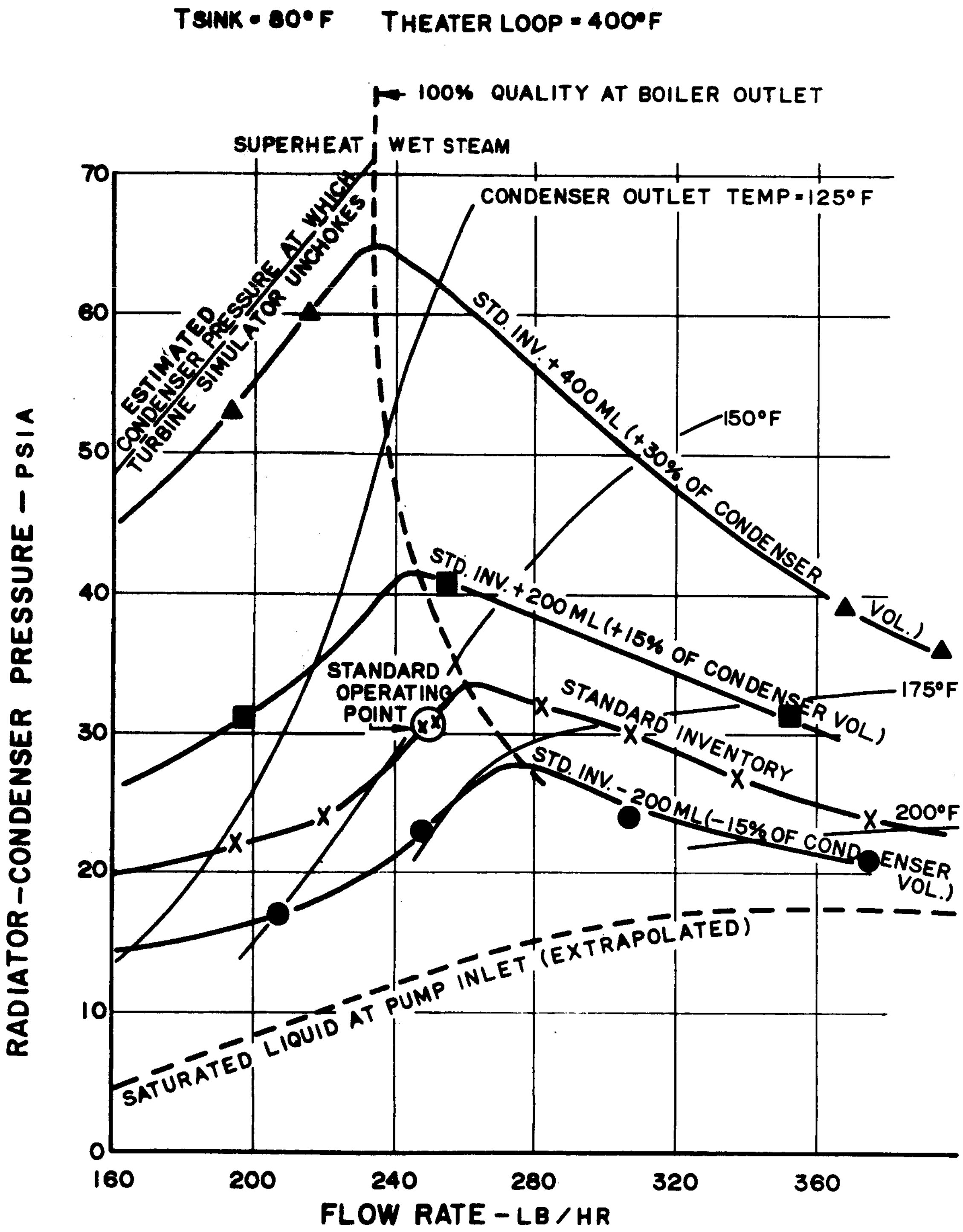

Figure 4 Steady-State Condensing Pressure vs Flow Rate at Different Fixed Total Loop Inventories. Constant Turbine Simulator Restriction (Choked)

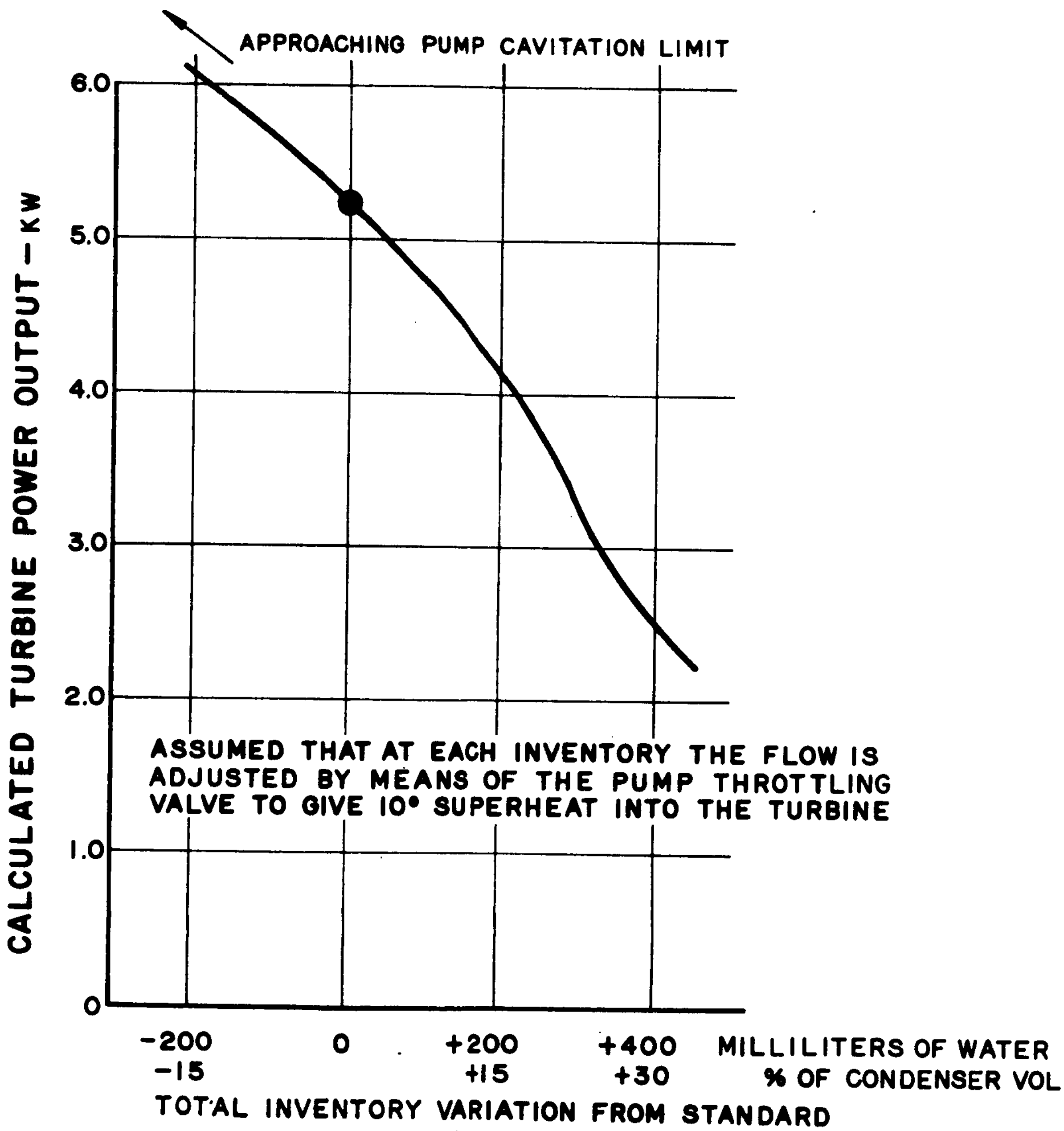

Figure 5 Calculated Turbine Power Output vs Inventory Variation. Calculations Based on Steady-State Data Taken with a Constant Turbine Simulator Restriction

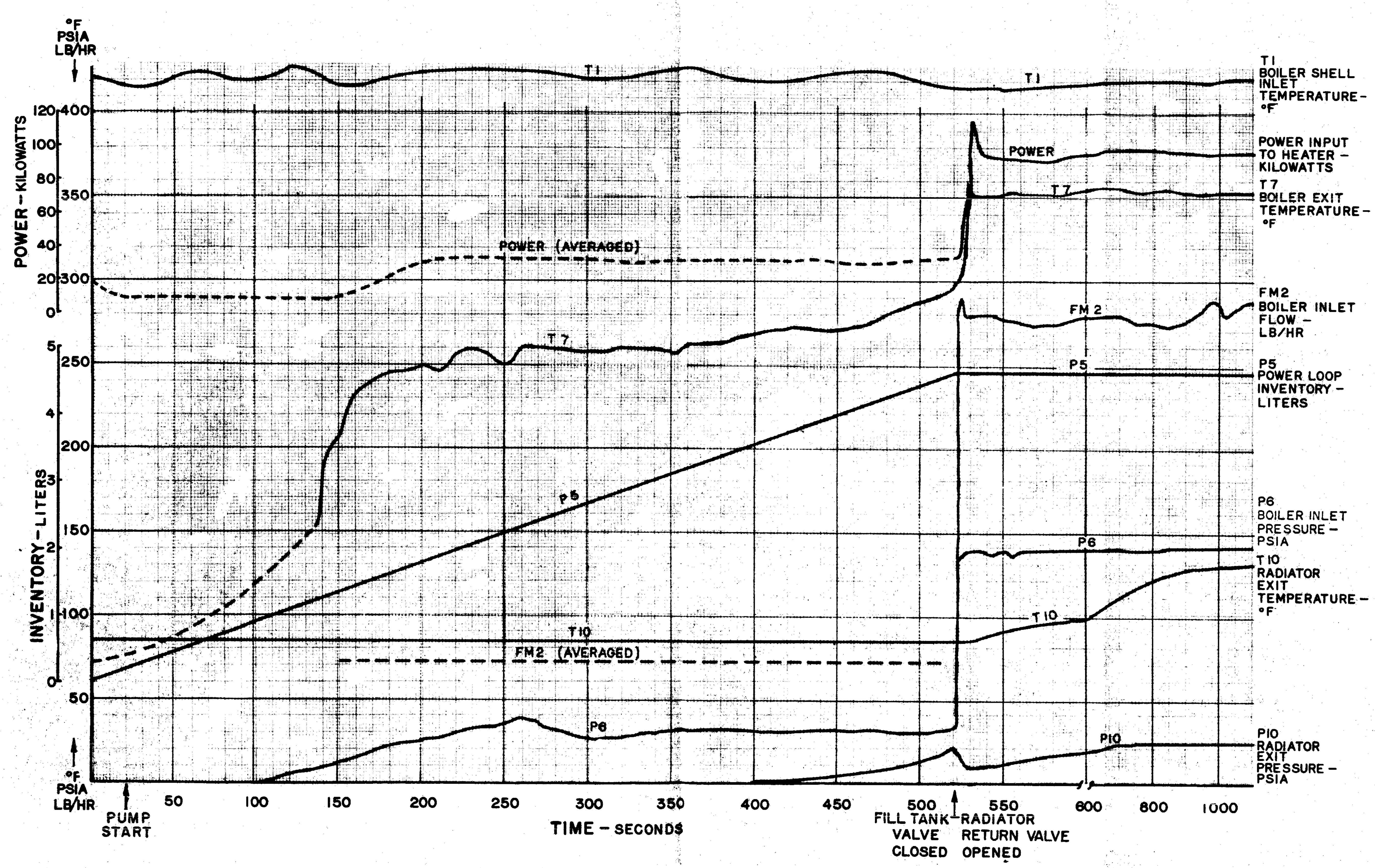

Figure 6 Standard Hard-Fill Starting Test

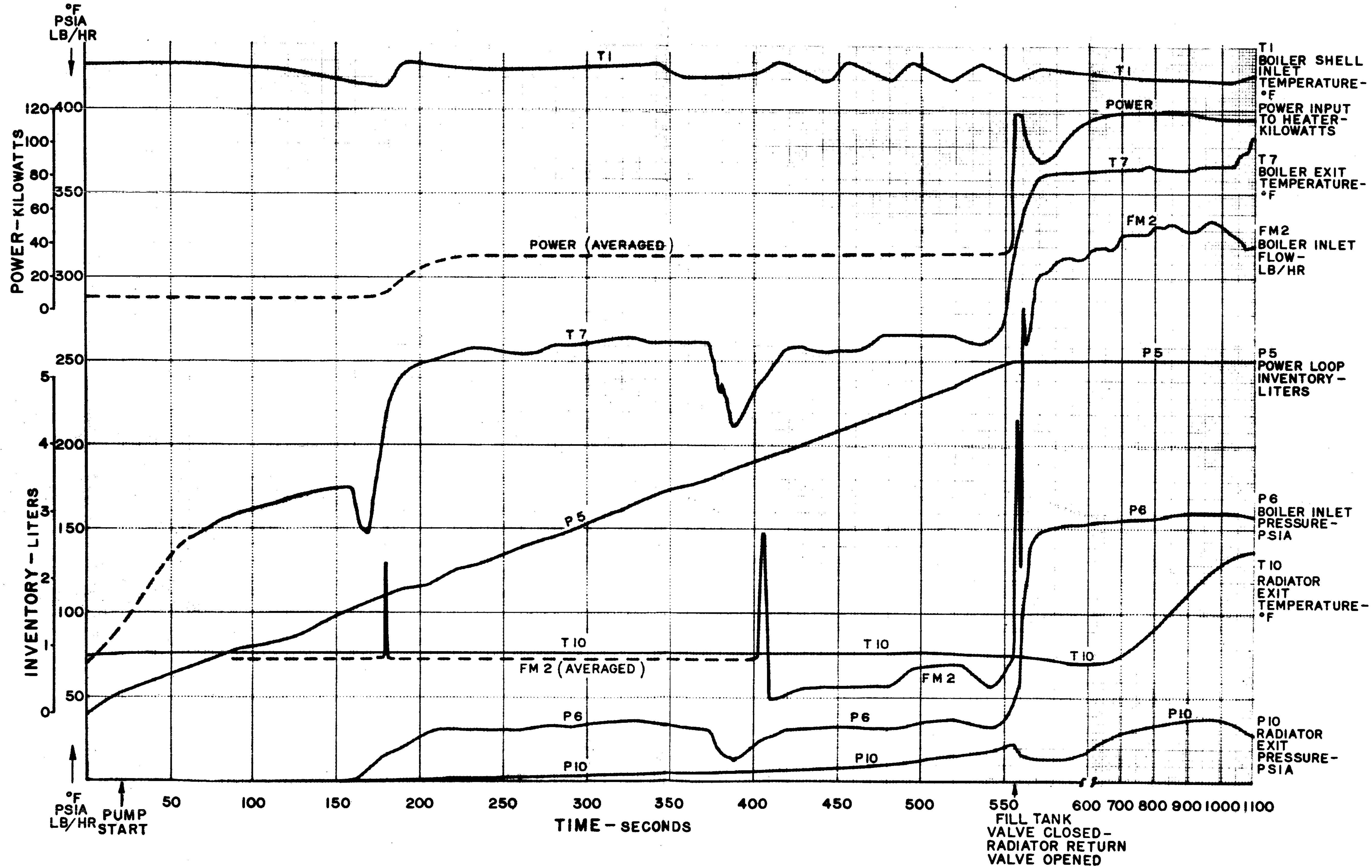

Figure 7 Hard-Fill Starting Test No. 3. Three Radiator Segments. Sink Temperature 60°F

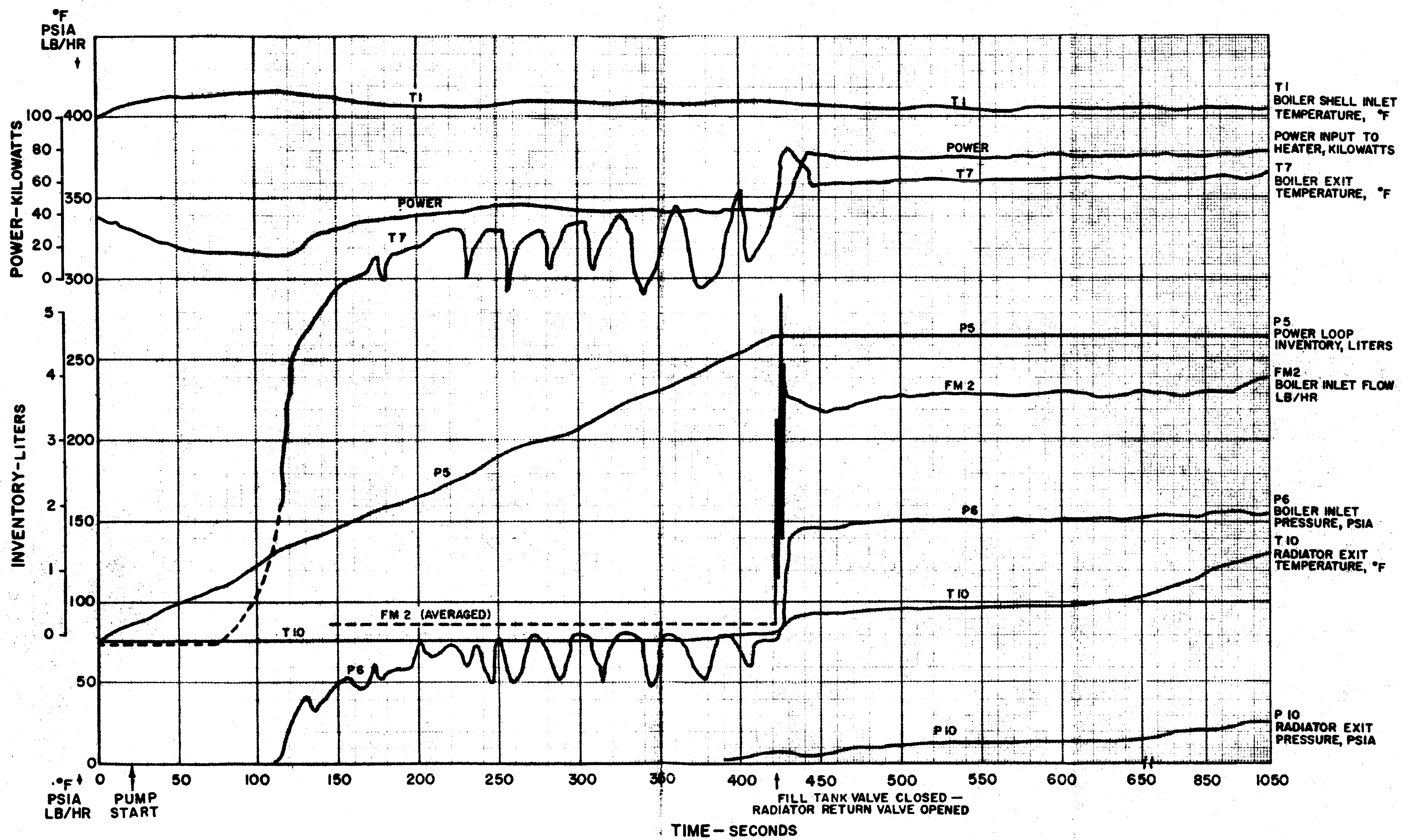

Figure 8 Hard-Fill Starting Test No. 4. Reduced Flow and Heater Temperature

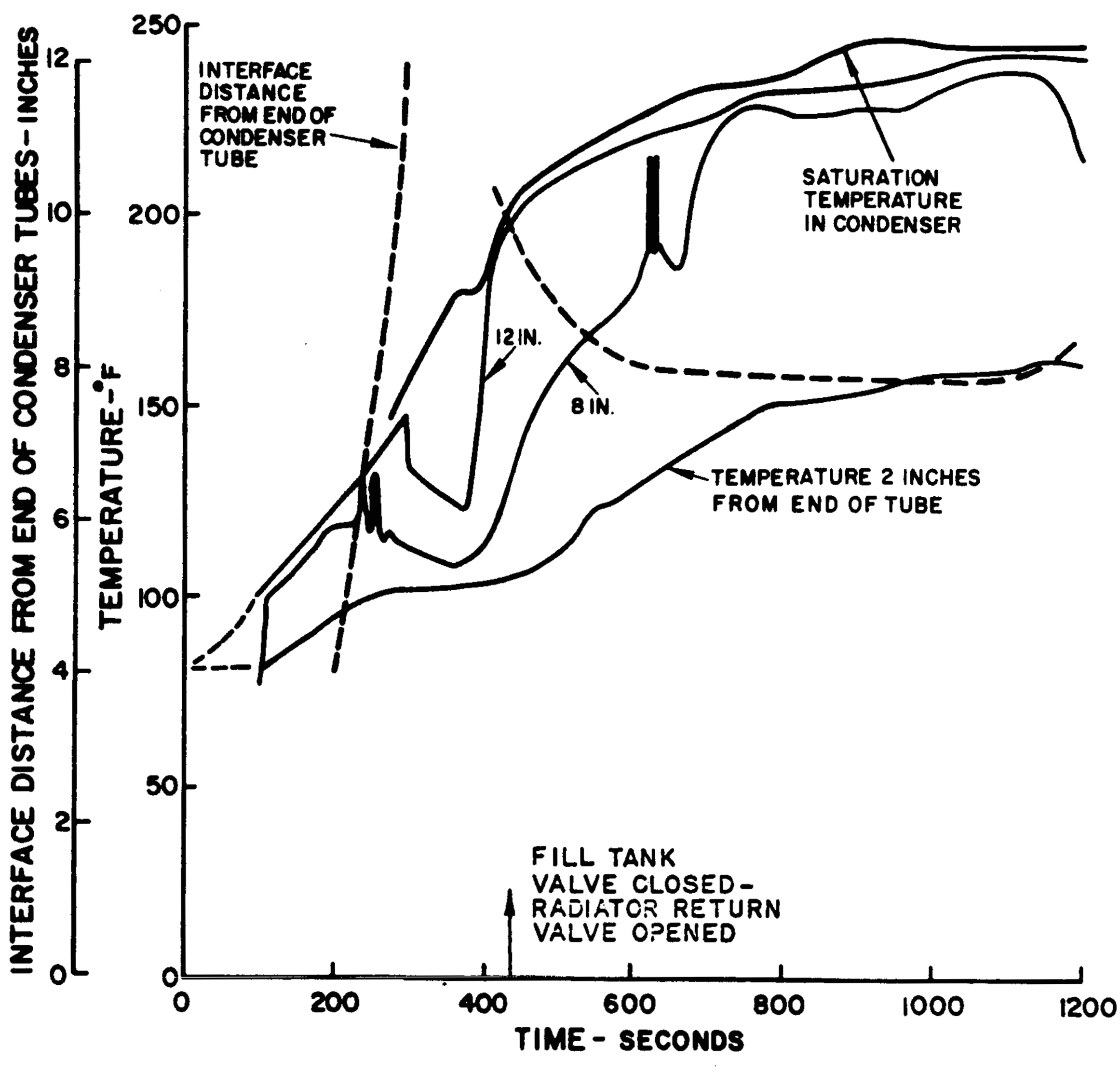

Figure 9 Hard-Fill Starting Test No. 4. Temperatures along Middle Tube of Radiator Segment A and Interface Location

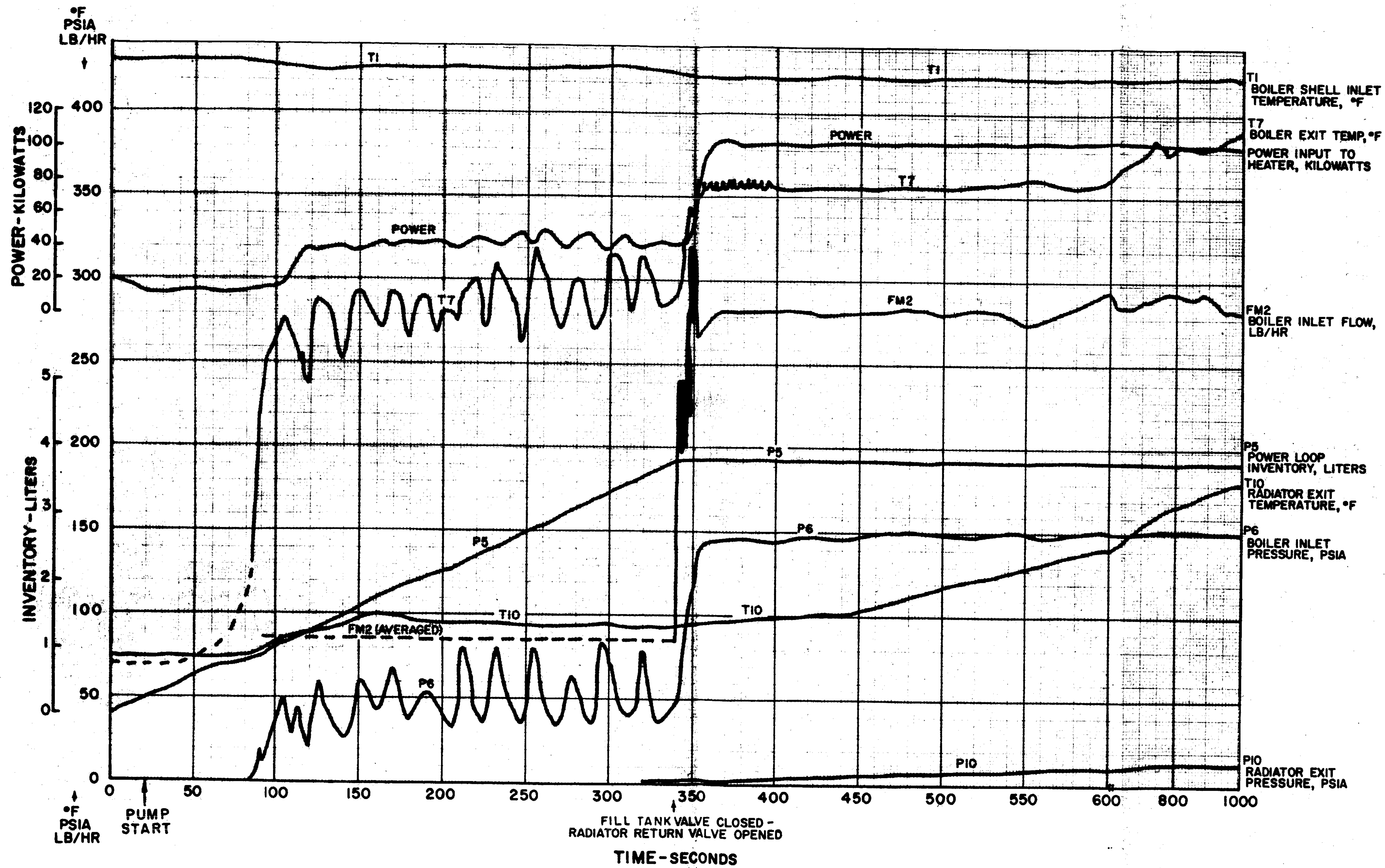

Figure 10 Hard-Fill Starting Test No. 5. Underfill by about 500 Milliliters

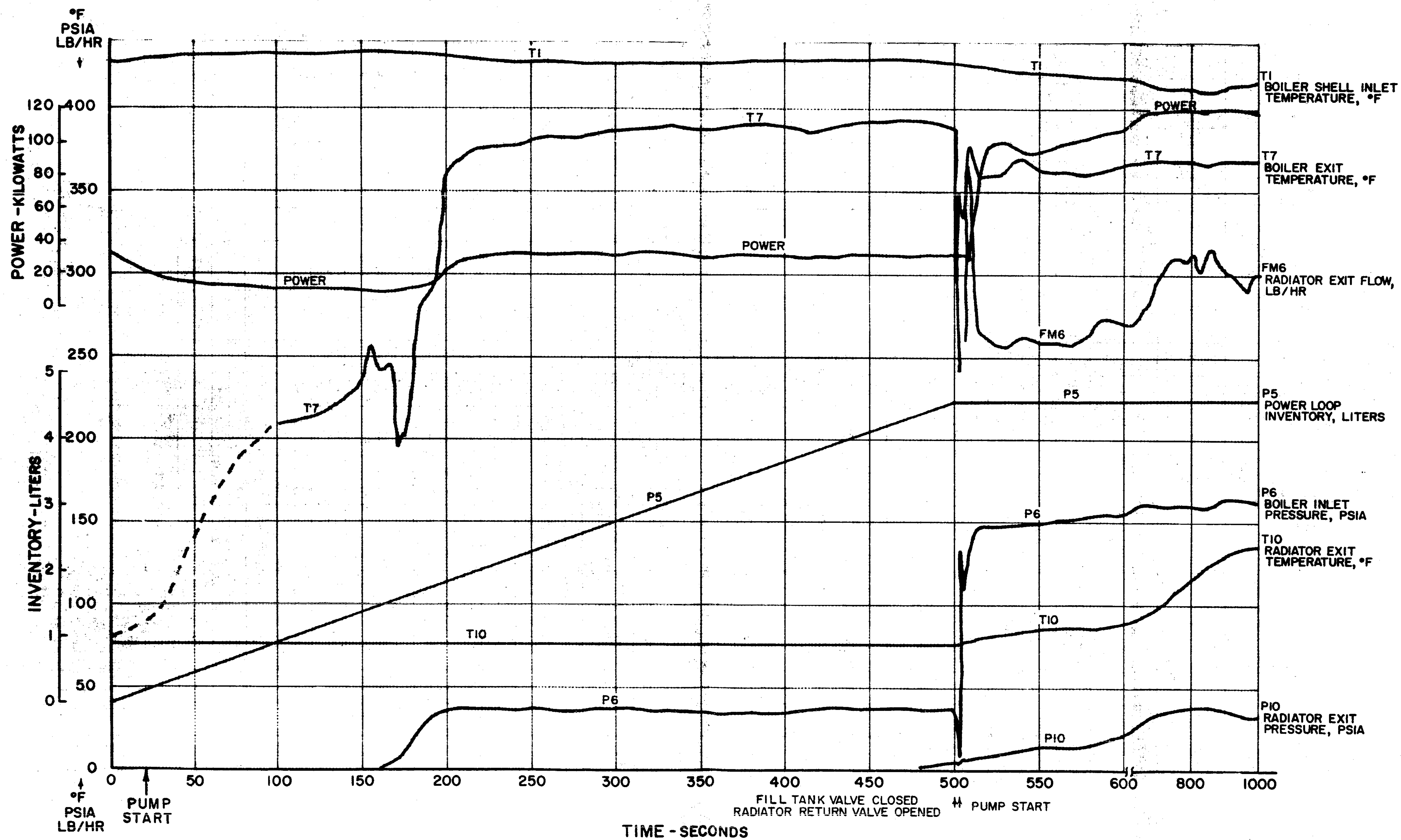

Figure 11 Hard-Fill Starting Test No. 6. Fill Tank at Pump Exit

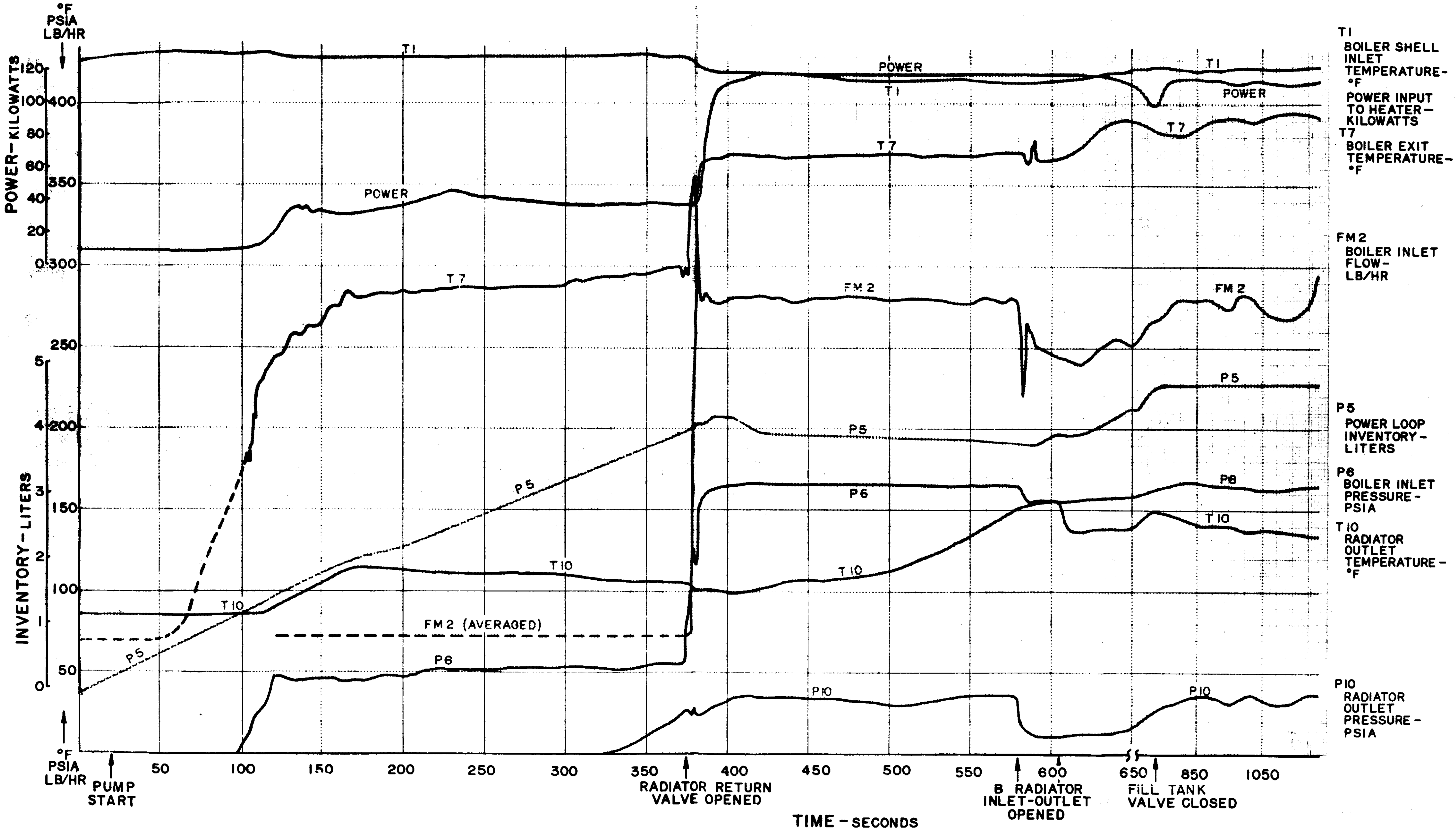

Figure 12 Hard-Fill Starting Test No. 7. Segmented Radiator

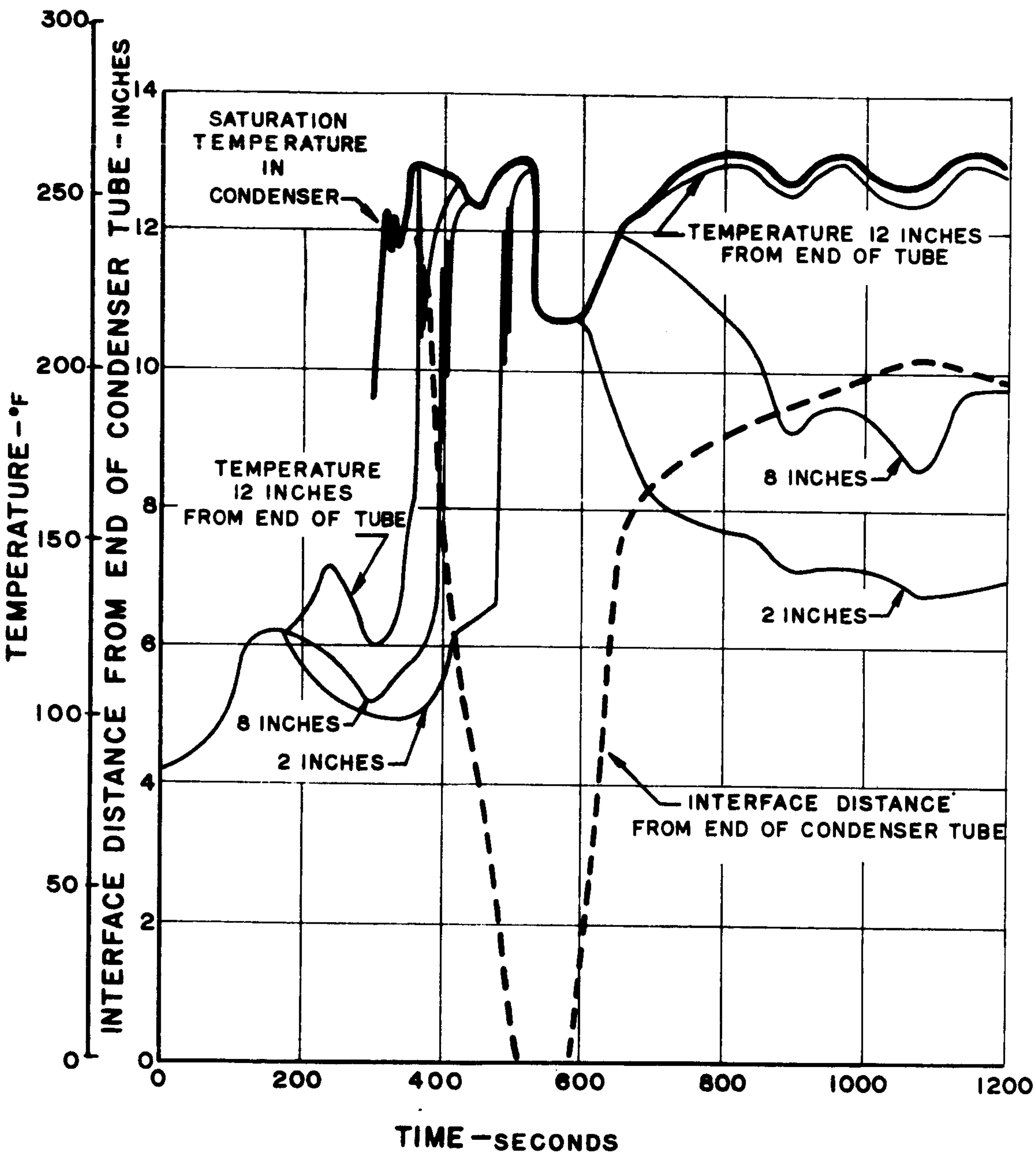

Figure 13 Hard-Fill Starting Test No. 7. Temperatures along Middle Tube of Radiator Segment A and Interface Location

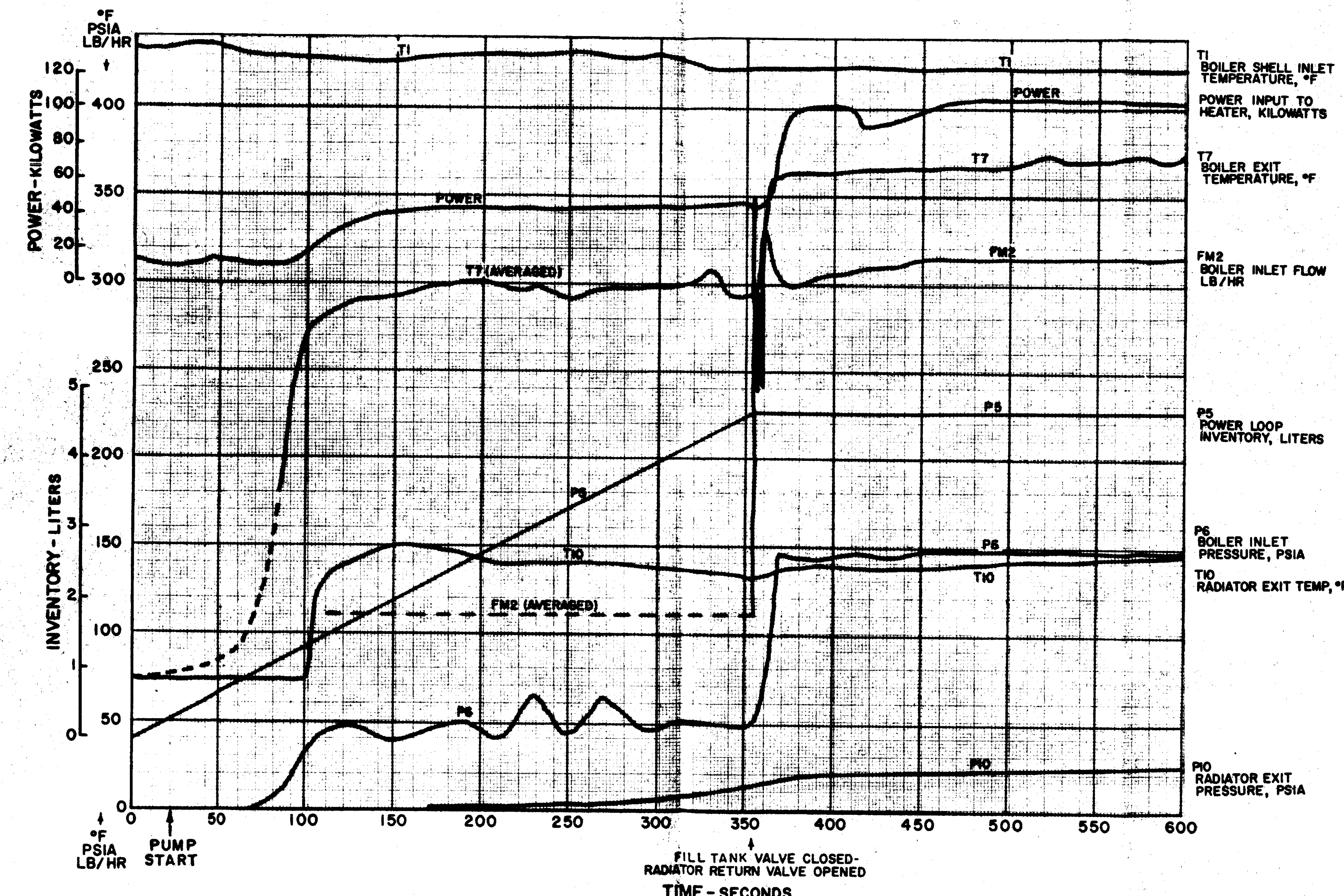

Figure 14 Hard-Fill Starting Test No. 8. Sink Temperature and Initial Radiator Temperature 140°F

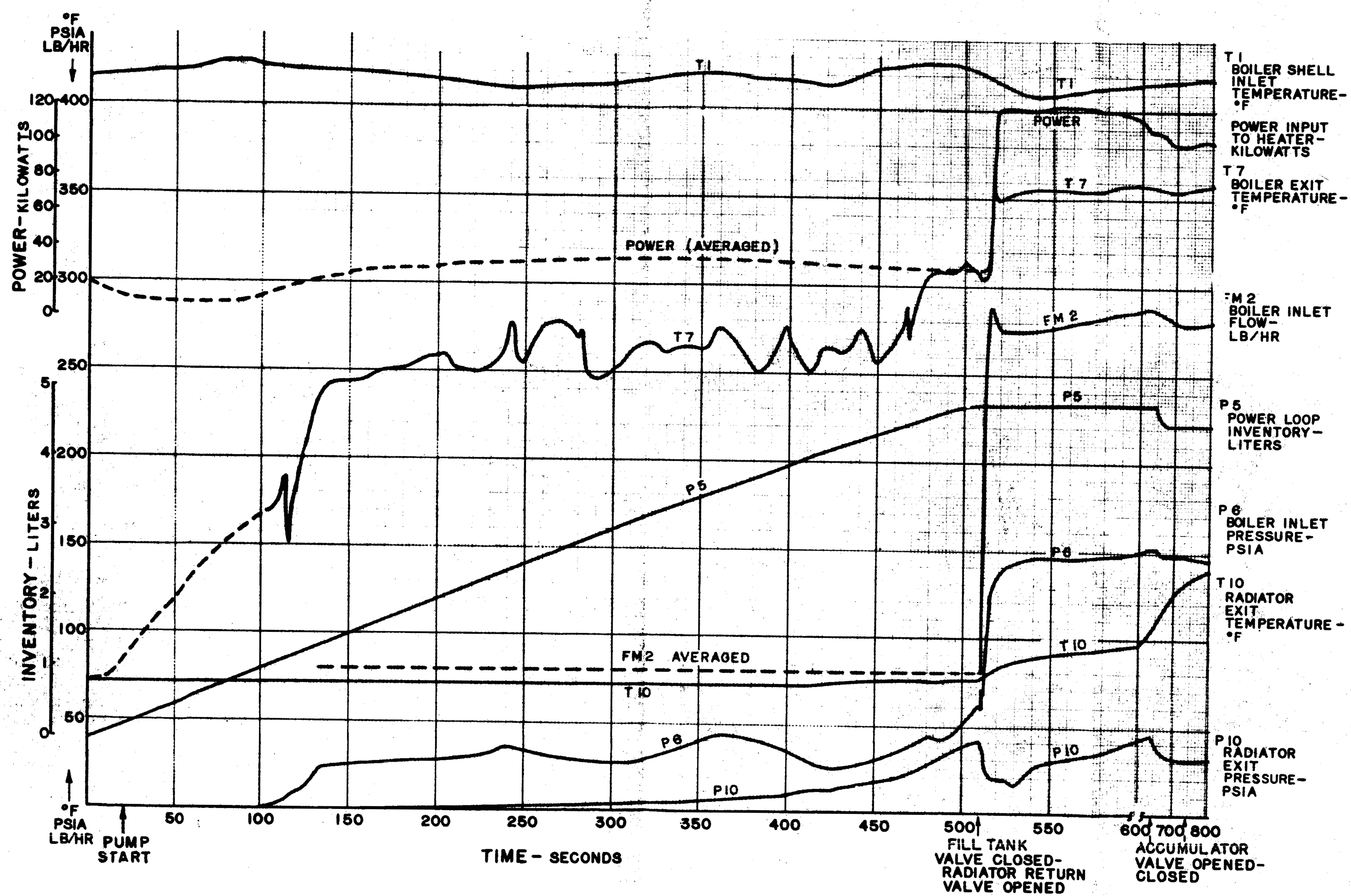

Figure 15 Hard-Fill Starting Test No. 9. Overfilled 0.25 Liter and Relieved

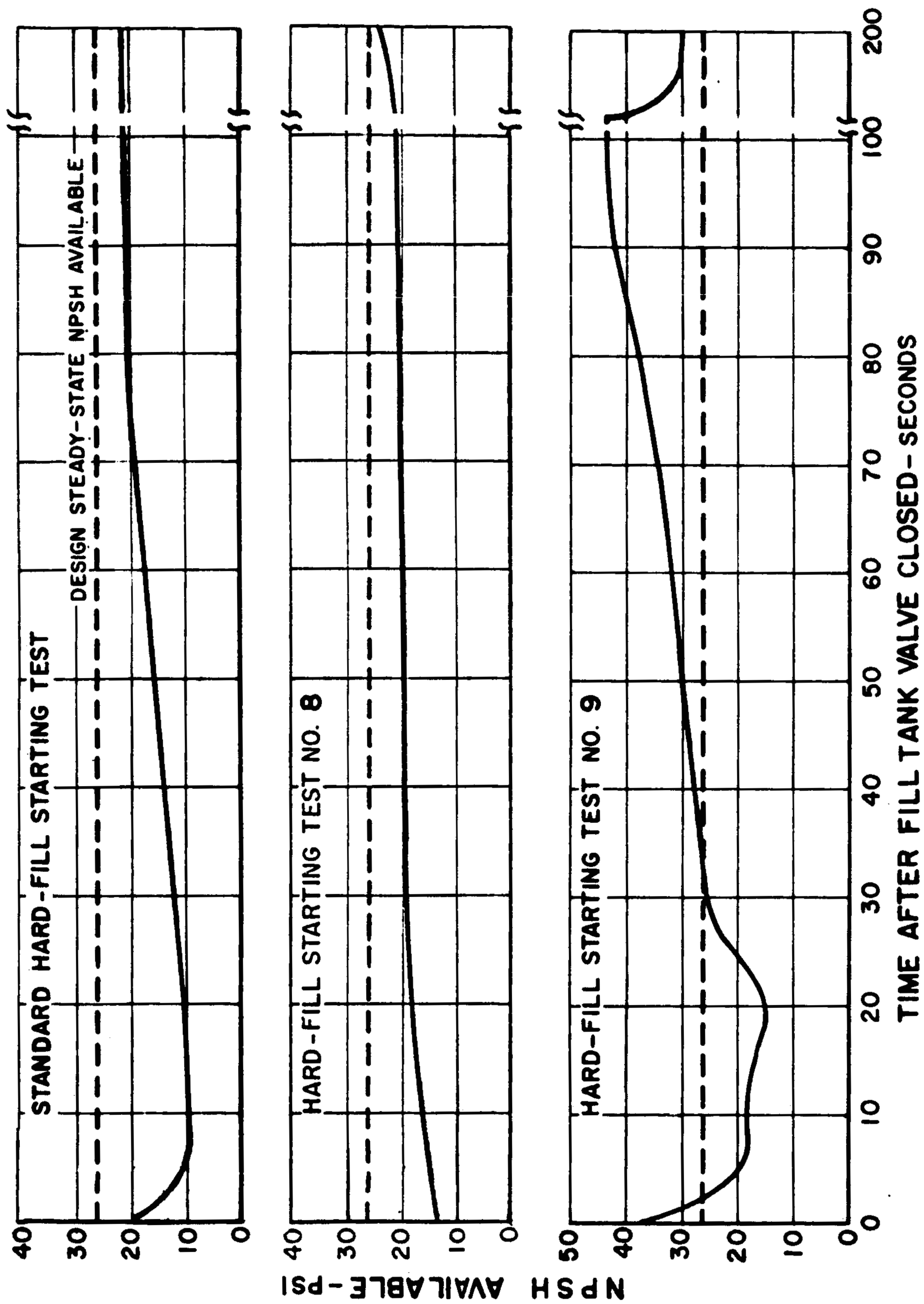

Figure 16 Net-Positive-Suction Head Available at Pump Inlet after Fill Tank Valve Closed. Standard Test and Tests Nos. 8 and 9

- POWER LOOP INITIALLY FILLED WITH AMBIENT TEMPERATURE WATER. ACCUMULATOR ATTACHED AT PUMP INLET WITH A RESTRICTION IN THE FILL LINE.
- THE POWER LOOP PUMP WAS STARTED (TIME = 0)
- THE HEATER WAS TURNED ON AT TIME = 0, AND THE HEATER LOOP TEMPERATURE INCREASED AT AN INITIAL RATE OF APPROXIMATELY 55°F/MIN.

OSCILLATIONS BEGAN WHEN T1 REACHED 275°F, APPROX. 3.5 MIN. AFTER START, AND PERSISTED UNTIL 7.8 MIN. (THE HEATER LOOP HAD STEADIED OUT AT 415°F). TIME OF RECORDING SHOWN IS APPROX. 7.2 MIN. AFTER TIME = 0

INSTRUMENT STATIONS

9-DOWNSTREAM OF TURBINE SIMULATOR VALVE
10-COMBINED RADIATOR OUTLET
12-POWER LOOP PUMP INLET
19-RADIATOR A OUTLET
20-RADIATOR B OUTLET
21-RADIATOR C OUTLET

T1-HEATER LOOP TEMPERATURE
T 117-END OF TUBE NO. 2, B SEGMENT
FM6-COMBINED RADIATOR OUTLET FLOW

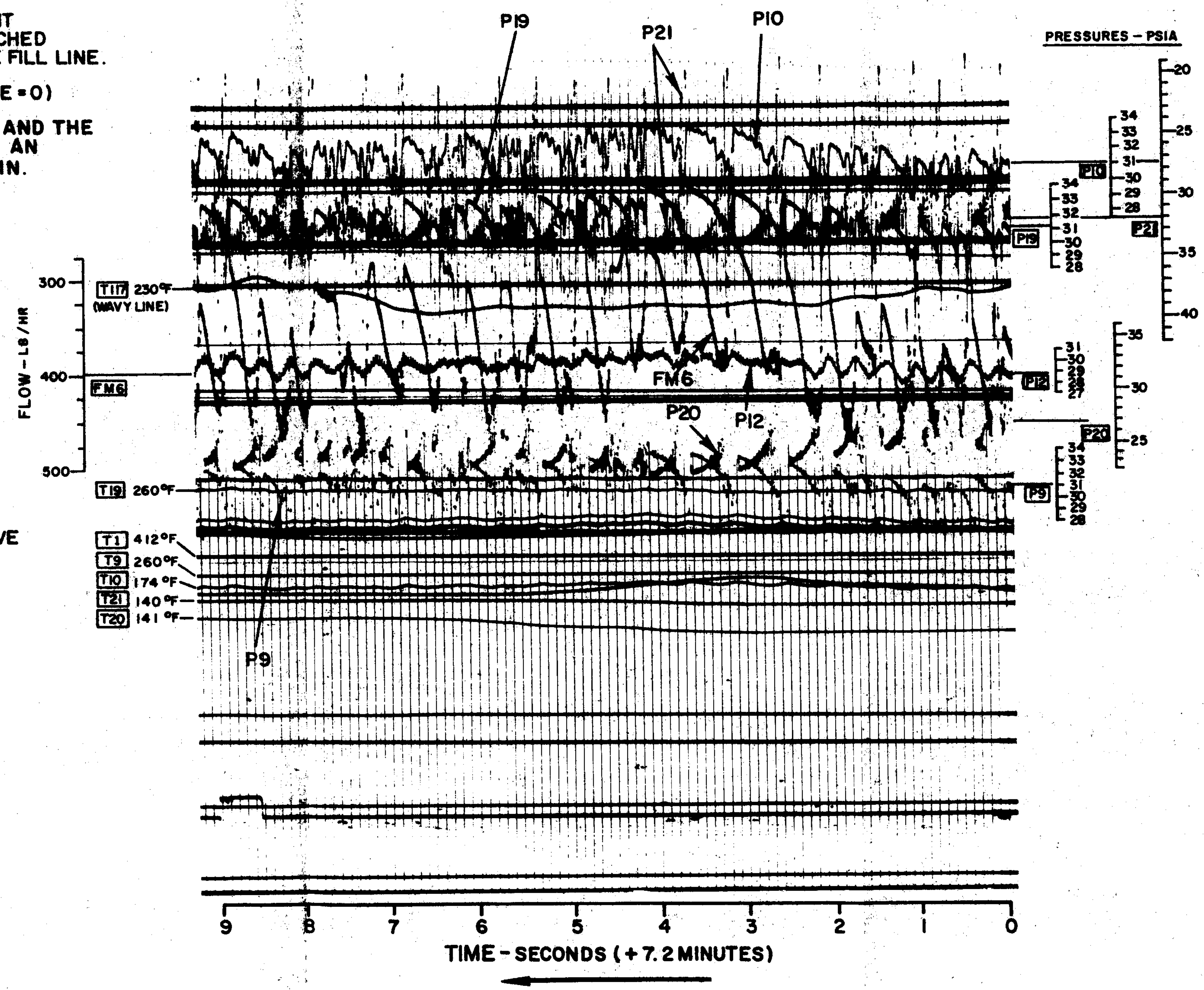

Figure 17 Oscillograph Recording Showing Instabilities Encountered during a Liquid-Filled Start

- POWER LOOP INITIALLY FILLED WITH AMBIENT TEMPERATURE WATER. ACCUMULATOR ATTACHED AT PUMP INLET WITH A RESTRICTION IN THE FILL LINE. THE HEATER LOOP WAS BROUGHT UP TO THE STEADY-STATE TEMPERATURE (415°F), AND THE VAPOR FORMED IN THE BOILER PUSHED LIQUID BACK INTO THE ACCUMULATOR.
- THE POWER LOOP PUMP WAS STARTED (TIME = 0)

INSTRUMENT STATIONS

6 - BOILER INLET
7 - BOILER OUTLET
9 - DOWNSTREAM OF TURBINE SIMULATOR VALVE
10 - COMBINED RADIATOR OUTLET
12 - POWER LOOP PUMP INLET
19 - RADIATOR A OUTLET
20 - RADIATOR B OUTLET
21 - RADIATOR C OUTLET

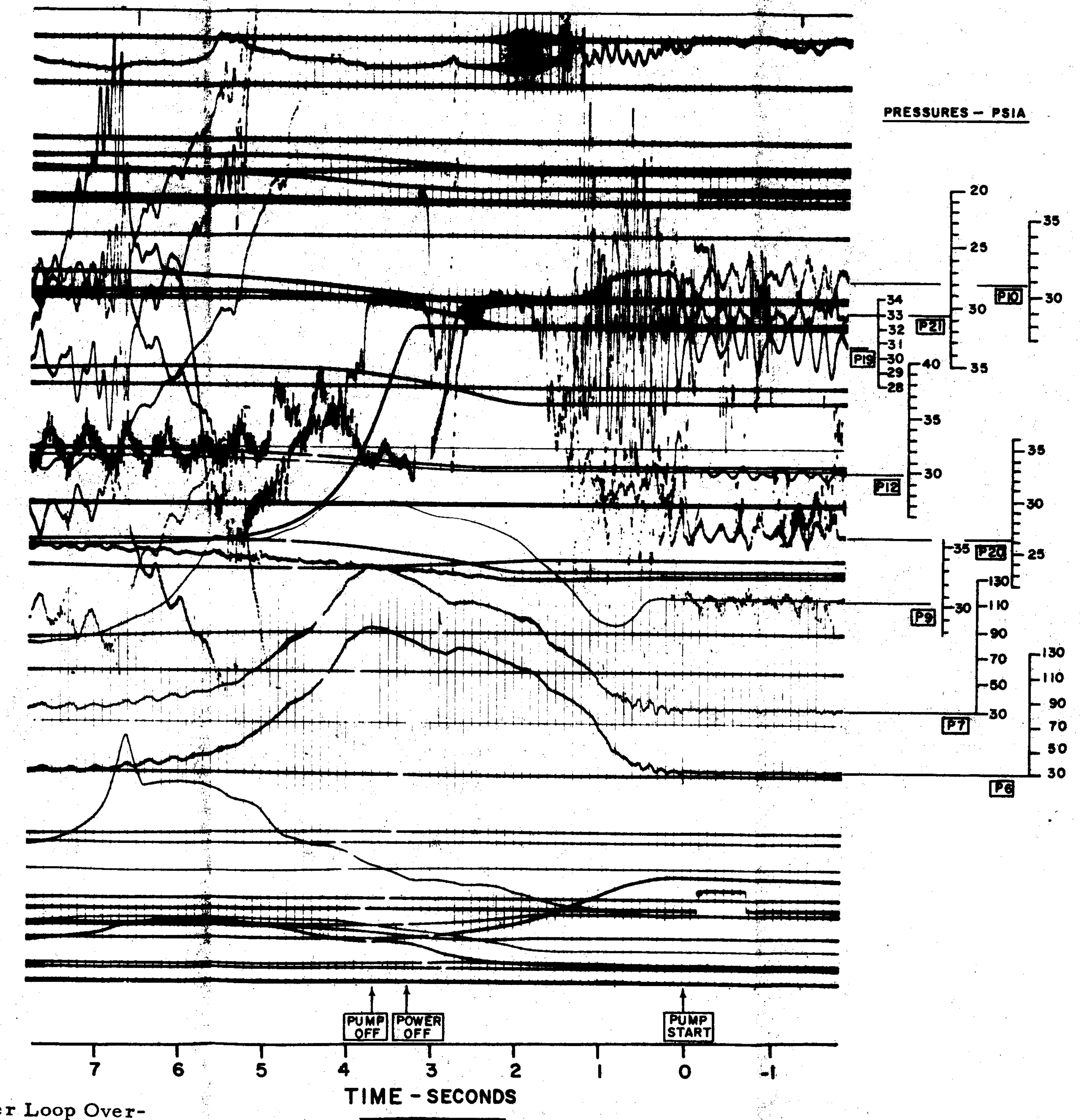

Figure 18 Oscillograph Recording Showing Power Loop Over-pressurization during a Liquid-Filled Start